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# TURBULENT CONVECTIVE HEAT TRANSFER IN AN ACOUSTICALLY RESONANT TUBE

A THESIS

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TURBULENT CONVECTIVE HEAT TRANSFER

IN AN ACOUSTICALLY RESONANT TUBE

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# NOMENCLATURE

## Latin

| | |
|---|---|
| $A$ | Inside surface area of tube, $ft^2$ |
| $C_p$ | Constant pressure specific heat, $Btu/lb_m$-$^oF$ |
| $D$ | Inside diameter of tube, ft |
| $G$ | Mass flow rate per unit area, $lb_m/hr$-$ft^2$ |
| $h$ | Convective heat transfer coefficient, $Btu/hr$-$ft^2$-$^oF$ |
| $h_{fg}$ | Latent heat of vaporization, $Btu/lb_m$ |
| $k$ | Thermal conductivity, $Btu/hr$-$ft$-$^oF$ |
| $L$ | Length, ft |
| $M$ | Mach number |
| $\dot{m}$ | Mass rate of flow, $lb_m/hr.$ |
| $p$ | Pressure, inches of $H_2O$ |
| $Q$ | Heat transfer, Btu |
| SPL | Sound pressure level, decibels (re 0.0002 $\mu$ bar) |
| $t$ | Temperature, $^oF$ |
| $V$ | Average air velocity, ft/sec |
| $x$ | Axial distance along tube, ft |

## Greek

| | |
|---|---|
| $\Delta$ | Increment |
| $\mu$ | Dynamic viscosity, $lb_m/ft$-$hr$ |
| $\nu$ | Kinematic viscosity, $ft^2/hr$ |
| $\rho$ | Density, $lb_m/ft^3$ |

| | |
|---|---|
| $\tau$ | Time, sec |
| $\Sigma$ | Summation |

Subscripts

| | |
|---|---|
| a | Refers to air |
| db | Dry bulb |
| f | Film temperature |
| lm | Logarithmic mean |
| wb | Wet bulb |
| i | Chamber number (1, 2, . . . 21) |
| m | Average, mean |
| o | Heat transfer entrance conditions |
| pc | Plenum chamber |
| s | Saturated steam |
| STAT | Static |
| $\infty$ | Asymptotic value |

Moduli

| | |
|---|---|
| Nu | Nusselt number based on tube diameter and film temperature properties ($hD/k_f$) |
| Re | Reynolds number based on tube diameter and inlet conditions ($GD/\mu_o$) |
| Gz | Graetz number |
| Gr | Grashof number |

## SUMMARY

This investigation was carried out to obtain quantitative heat transfer data for fully developed turbulent air flow at the entrance to a horizontal isothermal tube. The heat transfer data taken were in two phases, one with a resonant acoustic field impressed on the air flow and one without the resonant acoustic field. The purpose of taking data without the resonant acoustic field was to ascertain the reliability of the experimental apparatus by comparing the data taken with that found in the literature.

The experimental work and empirical equations obtained by Latzko (1) for fully developed turbulent flow were chosen as the standard for comparison and the agreement was found to be excellent. Latzko's empirical equations for the asymptotic and average heat transfer coefficient are

$$h_\infty = \frac{(0.23)(c_p)(G)}{(DG/\mu_f)^{0.20}(\mu_f c_p/k_f)^{0.67}} \tag{1}$$

and

$$\frac{h_m}{h_\infty} = 1 + \frac{1.4\,D}{L}\ ,\ \left(\frac{L}{D} > 5\right) \tag{2}$$

The value of the average heat transfer coefficient using the above equations was within $\pm$ 2 per cent of the measured average heat transfer coefficient.

The range of Reynolds numbers investigated was 9,900 to 46,500 with sound pressure levels of up to 154 db (154 db was the maximum obtainable sound pressure level with the sound generating equipment used). The effect of the resonant acoustic field on the flow was determined by comparing the no-sound runs with those runs having the resonant acoustic field present at the same Reynolds number.

No effect on heat transfer by the sound field in the range of Reynolds numbers between 15,800 and 46,500 could be detected. Therefore, if a threshold sound pressure level exists in this range of Reynolds numbers, it is above 154 db. The threshold sound pressure level was exceeded at a Reynolds number of 9,900, but insufficient data were taken to determine the threshold sound pressure level as a function of the Reynolds number.

# CHAPTER I

# INTRODUCTION

## Background

Interest in the effects of acoustic vibrations on convective heat transfer has been stimulated in recent years by the failure of tailpipes on turbo-jet and rocket engines. It has been shown that acoustic vibrations influence the local convective heat transfer coefficient.

A program to study the effects of acoustic vibrations on heat transfer was begun at the Mechanical Engineering Laboratories of the Georgia Institute of Technology in 1958, under the sponsorship of the Aeronautical Research Laboratory at Wright Field.

An apparatus that consisted essentially of a horizontal copper tube enclosed in a steam chest was constructed to measure heat transfer coefficients for air flowing in the tube. A blower and air plenum chamber with a bell mouth entrance provided air with a developing velocity profile at the thermal entrance of the horizontal copper tube. Various support equipment and instrumentation were used with the basic apparatus.

Investigations (3, 4) revealed the existence of critical or threshold sound pressure levels in the horizontal isothermal tube, levels below which there were no observed effects on heat transfer due to resonant acoustic vibrations. These critical or threshold values were shown to be a function of the Reynolds number, and the data reported were for developing flow at the entrance to the isothermal test section.

Eastwood's (1) investigation with this apparatus showed two regions of interest in which the heat transfer characteristics were markedly different. Below a Reynolds number of approximately 40,000 the resonant sound field caused the local Nusselt number to vary periodically about the no-sound value of the Nusselt number as shown in Figure 1, while above a Reynolds number of approximately 40,000 the local Nusselt number was shown to vary periodically. However, in the latter case the value of the local Nusselt number was always below that of the no-sound case as shown in Figure 2. Eastwood also showed that critical or threshold values of the sound pressure level existed and that the threshold value was a function of the Reynolds number. Figure 3 shows the threshold sound pressure level as a function of Reynolds numbers between 16,000 and 200,000 with a maximum sound pressure level of 164.5 db at a resonant frequency of 222 cps. It was shown that the local heat transfer coefficient varied periodically between the nodes and loops of the resonant sound wave. Below a Reynolds number of 40,000 the maximum local Nusselt number occurred at the velocity loops, and above 40,000 the maximum values of the Nusselt number shifted to the velocity nodes. A pronounced change in the threshold sound pressure level was evident at this shift as seen in Figure 3. No physical explanation has been given for this shift, but Eastwood showed that the break in the threshold sound pressure level occurred near the point where the maximum acoustic velocity was equal to the mean through-flow velocity.

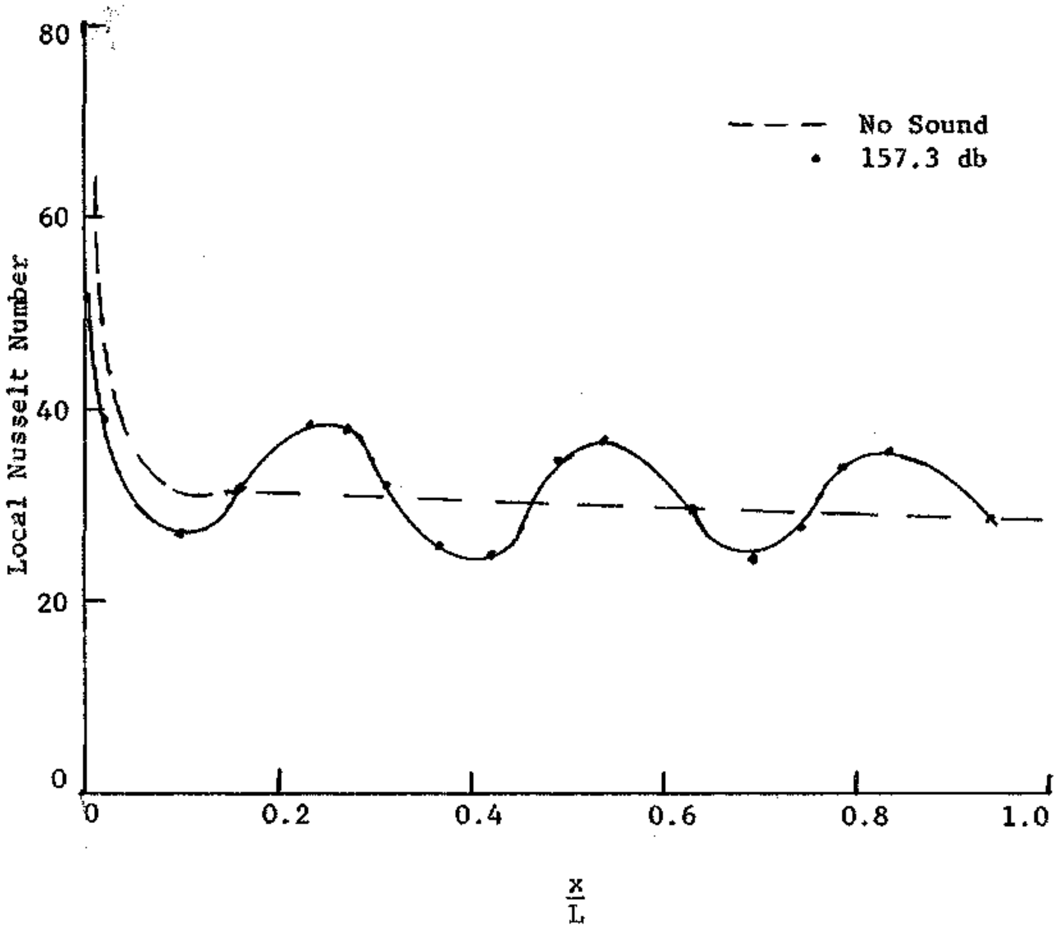


Figure 1. Local Nusselt Number vs. $\frac{x}{L}$ Re = 11,600

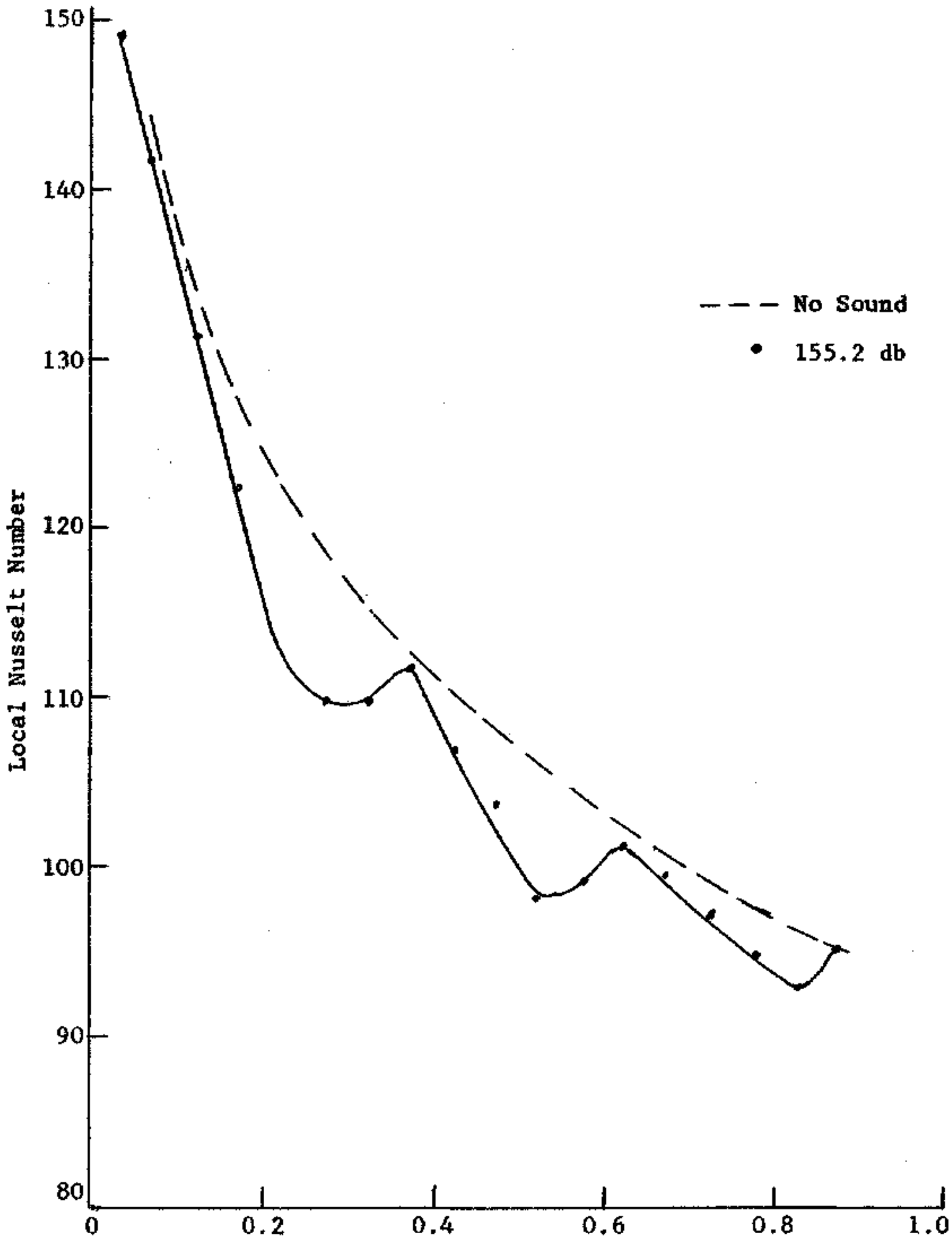


Figure 2. Local Nusselt Number vs. $\frac{x}{L}$ Re = 43000

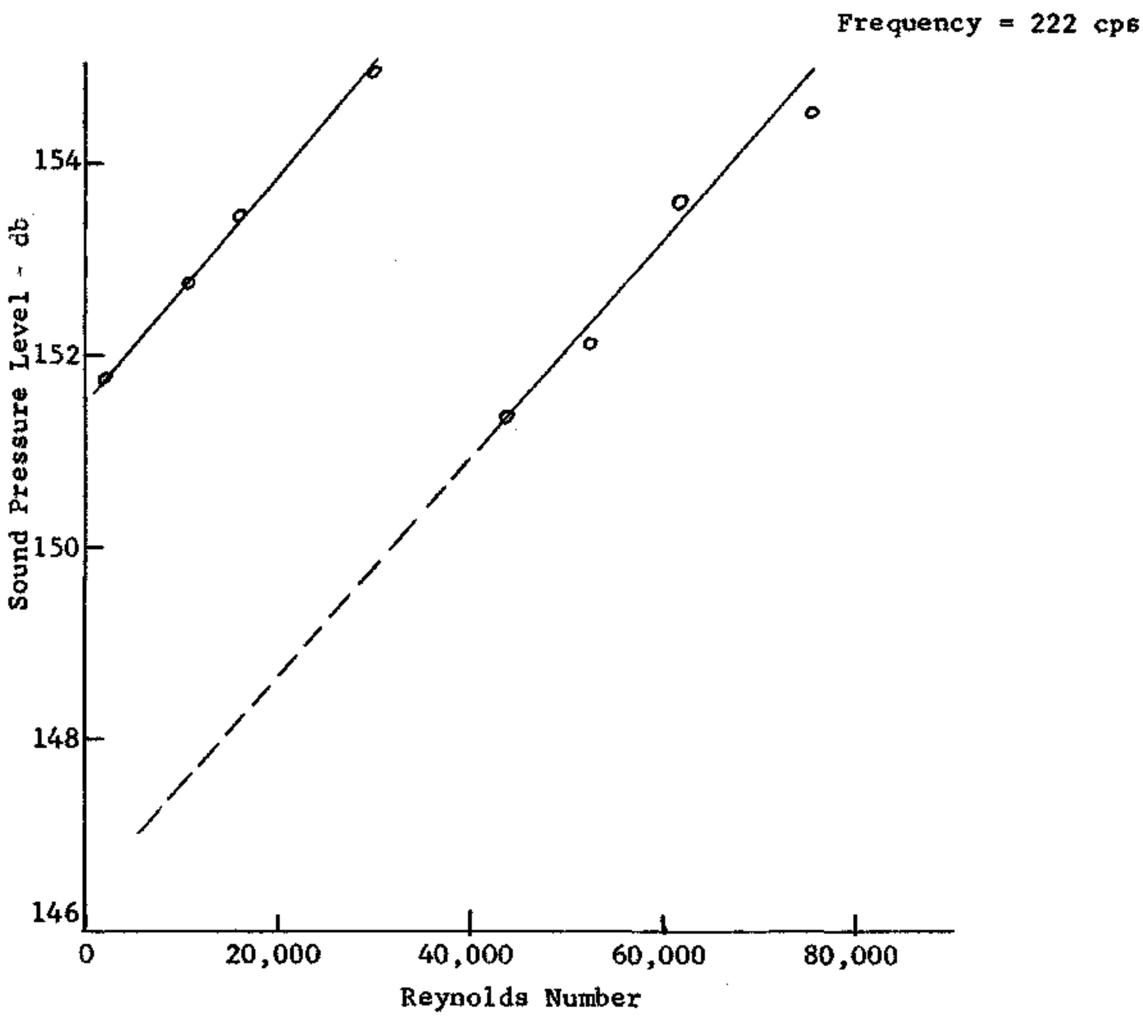


Figure 3. Threshold SPL for Developing Flow

Purdy (4), using Eastwood's data, showed that the threshold SPL in the range of Reynolds numbers between approximately 10,000 and 35,000 can be described by the empirical equation

$$\frac{M}{M_o^2} = 63$$

where

$M$ = Mach number based on the through-flow velocity.

$M_o$ = Mach number based on the maximum particle velocity due to a resonant sound field.

He also pointed out that the reason this equation was not valid below Re = 10,000 was that the effects of combined free and forced convection were appreciable below that Reynolds number. Heat transfer tests without sound verified this.

## Purpose and Scope

There were three main objectives in this investigation:

1. To construct an apparatus that was reliable and would provide heat transfer data consistent with known data.

2. To determine if a threshold sound pressure level for fully-developed turbulent air flow existed in the range of Reynolds numbers between approximately 10,000 and 50,000.

3. To compare the threshold sound pressure level for fully developed flow (if it existed) with that for developing flow determined by Eastwood (1), and, if they were similar, determine if the curve for

threshold sound pressure level at higher Reynolds numbers could be extrapolated to lower Reynolds numbers (see dotted line on Figure 3). The experimental work of Latzko (2) was chosen as the standard for comparison in 1 above.

All data and results are entirely experimental in nature. The controllable variables in this investigation were air flow rate, air temperature, sound pressure level (SPL), and frequency. The range of Reynolds numbers investigated was $9{,}910 < Re < 46{,}501$ ($.0054 < M < .0200$) with sound pressure levels up to 154 db.

# CHAPTER II

# APPARATUS

## Description

Figure 4 is a schematic drawing of the experimental apparatus used in this investigation. The apparatus consisted of:

1. Air Circulating System
    a. Blower
    b. Mixing compartment
    c. Laminar flow meter
    d. Micro-manometer
    e. Heater
    f. Plenum chamber
    g. Hydrodynamic entrance section
    h. Return duct
    i. Throttle
2. Heat Transfer Section
    a. Steam heated tube and condensate collectors
    b. Thermocouples
3. Sound Generating Equipment
    a. Drivers and horn
    b. Signal generator and amplifier
4. Sound Measuring Equipment
    a. Microphone
    b. SPL meter

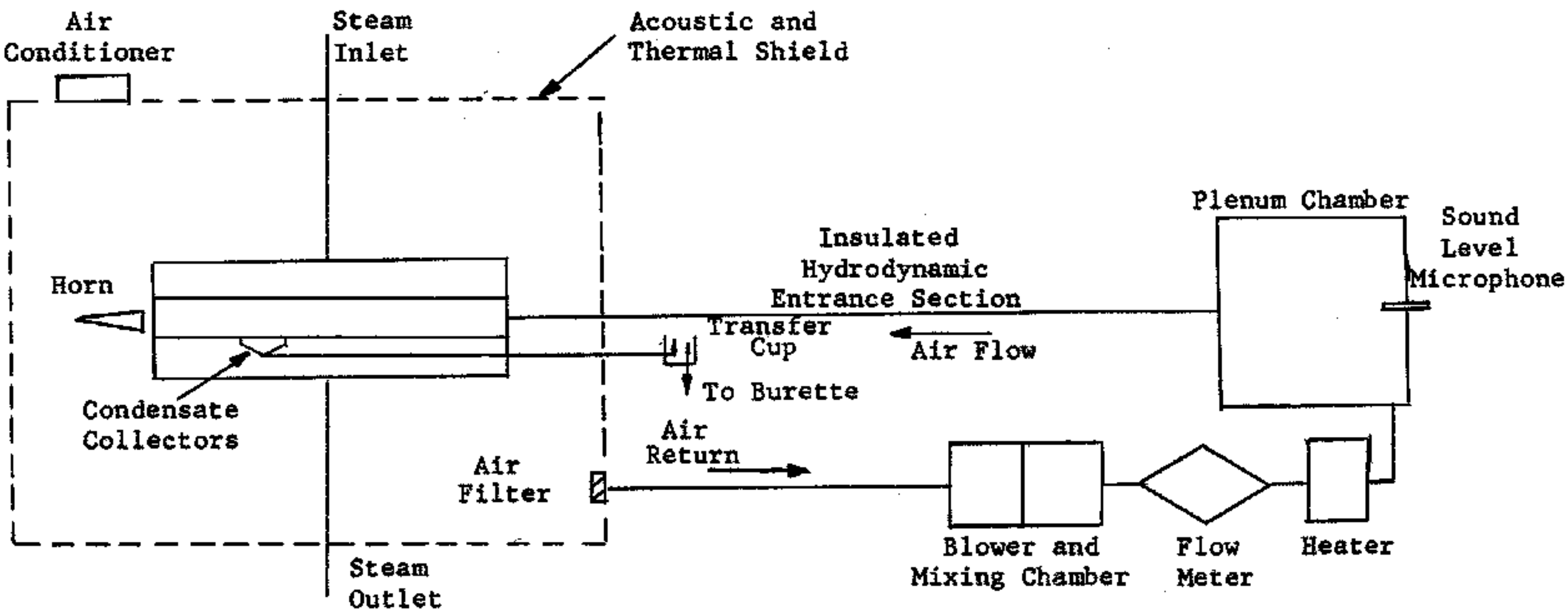


Figure 4. Schematic Diagram of Experimental Apparatus

5. Thermal and Acoustic Shield

6. Support Instrumentation

Detailed description of components of apparatus:

1. <u>Air Circulating System (See Figure 5)</u>.

a. Blower. The blower used for circulating the air was a Lockwood pressure blower Model PB-10 with a capacity of 360 cfm at seven inches of water pressure. The blower provided a steady flow of air with negligible pulsation.

b. Mixing compartment. The mixing compartment was a hollow box 22 by 22 by 25 inches made from half-inch plywood. Here the air was mixed after being discharged from the blower.

c. Laminar flow meter. This was a Meriam Instrument Company Model 50MC2-4P flow measuring instrument. The device indicated volumetric flow rate by producing a differential pressure across a flow element that was measured with a micro-manometer. It had a linear response between volumetric flow rate and differential pressure with a flow range of 100 to 1 possible. The instrument was accurate to $\pm$ 0.5 per cent of The Meriam Instrument Company flow standards.

d. Micro-manometer. A Meriam Instrument Company Model 34FB2 micro-manometer was used for measuring the static pressure and the differential pressure at the laminar flow meter. Accuracy of $\pm$ 0.001 inches of water pressure was possible with a range of differential pressure of ten inches of water. The instrument had a maximum operating pressure of 20 psig. All wetted parts were made of stainless steel.

e. Heater. An electrical resistance heater was placed in the air stream immediately downstream from the laminar flow meter. It

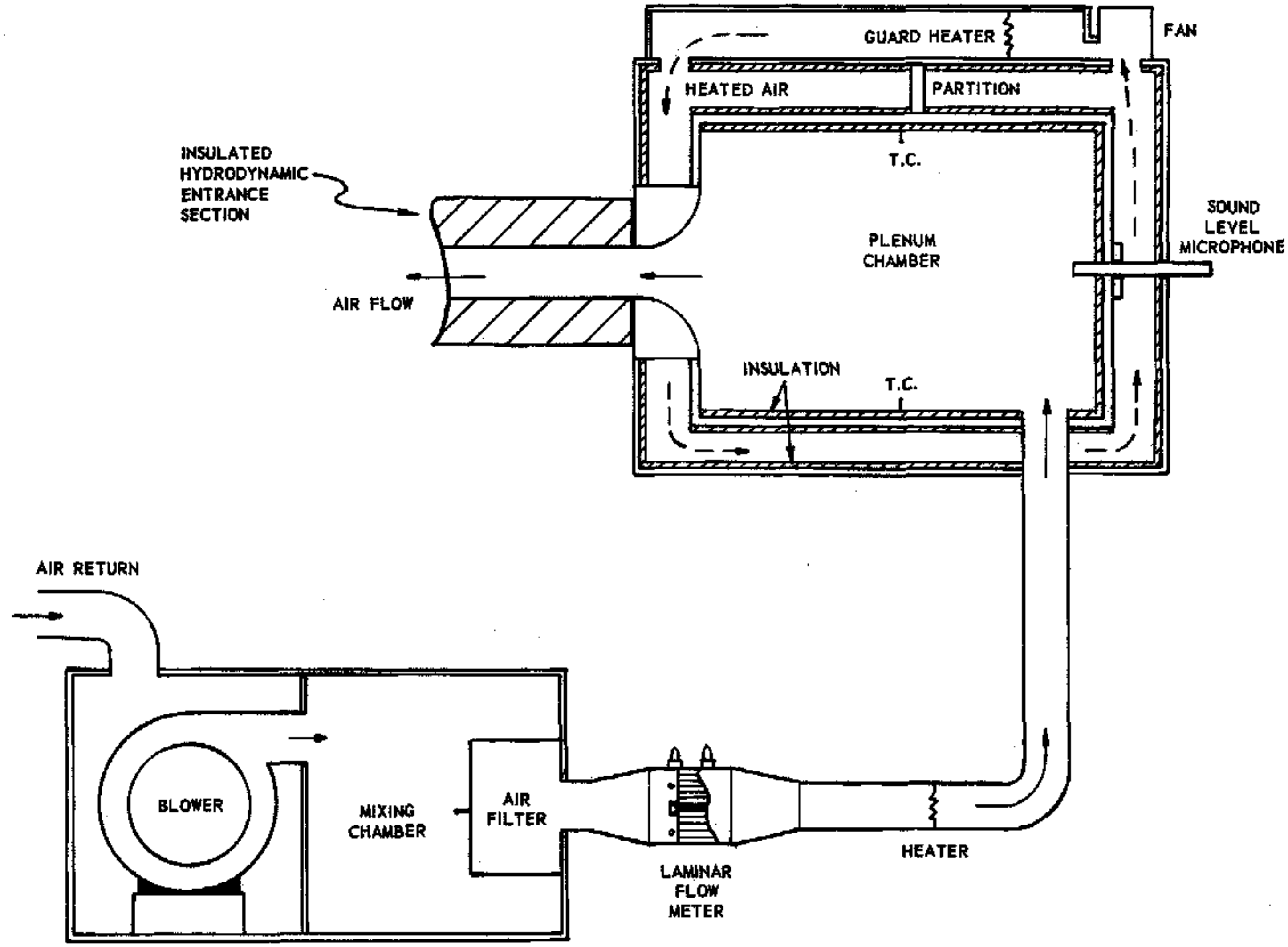


Figure 5. Air Circulation System.

was used to hold the air temperature constant within the plenum chamber. Controlled automatically with a Leeds & Northrup Type G Series 60 Speedmax Recorder in conjunction with a controller, the heater held the air temperature within the plenum chamber at the desired level to within $\pm$ 0.25°F. Four 24-gauge copper-Constantan thermocouples connected in parallel were placed in the plenum chamber and connected to the recorder. The recorder monitored the thermocouples and automatically controlled the air heater. An air temperature of 110°F was chosen in this investigation.

f. Plenum chamber. The plenum chamber consisted of two plywood boxes, one within the other (see Figure 5). The inner chamber was fabricated from three-quarter inch plywood and was insulated with one-inch Johns-Manville fiber glass on both the inside and outside surfaces. The outer box was made from one-half inch plywood and it was insulated only on the inside surface with one-inch Johns-Manville fiber glass insulation. When the two boxes were assembled, there was a two-inch air gap between them. The air within the gap was circulated by a small fan and heated by a guard heater to the same temperature as the air within the inlet plenum chamber. This eliminated heat transfer to or from the plenum chamber. Two thermocouples were placed in the air gap and connected to a Leeds & Northrup Type H direct reading temperature indicator. These temperatures were monitored visually and the guard heater was controlled manually.

g. Hydrodynamic entrance section. The insulated hydrodynamic entrance section was a 3.86 inch I.D., 28.26 foot long copper tube insulated with two inches of Johns Manville "Micro-Lox." This section was approximately eighty-five diameters long and was used to fully develop the velocity profile of the air before it entered the heated section. The temperature drop of the air from the plenum chamber to the heated tube in the hydrodynamic entrance was approximately $1^{o}F$ for a Reynolds number of 10,000. This temperature drop in the entrance section caused at most a 2.5 per cent error in the local heat transfer coefficient. For higher Reynolds numbers the temperature drop in the entrance section was less than $1^{o}F$ and a smaller error resulted. No attempt was made to apply a correction factor for this temperature drop since this error caused only a small variation in the average heat transfer coefficient, and because the error was smaller at higher Reynolds numbers.

h. Return duct. The return duct was a four-inch diameter galvanized pipe used for returning air from the heat transfer section to the blower.

i. Throttle. The throttle (a six-inch gate valve) was used to control the flow rate of the air.

2. Heat Transfer Section (See Figure 6).

a. Steam heated tube and condensate collectors. The steam heated tube was a 3.86 inch I.D., 10.5 foot long copper tube surrounded by a 16-inch standard steel pipe forming an annular steam chest around the copper tube. Saturated steam was allowed to circulate within the steam chest to keep the copper tube at isothermal conditions. Condensate collectors were fastened to the bottom of the copper tube at evenly

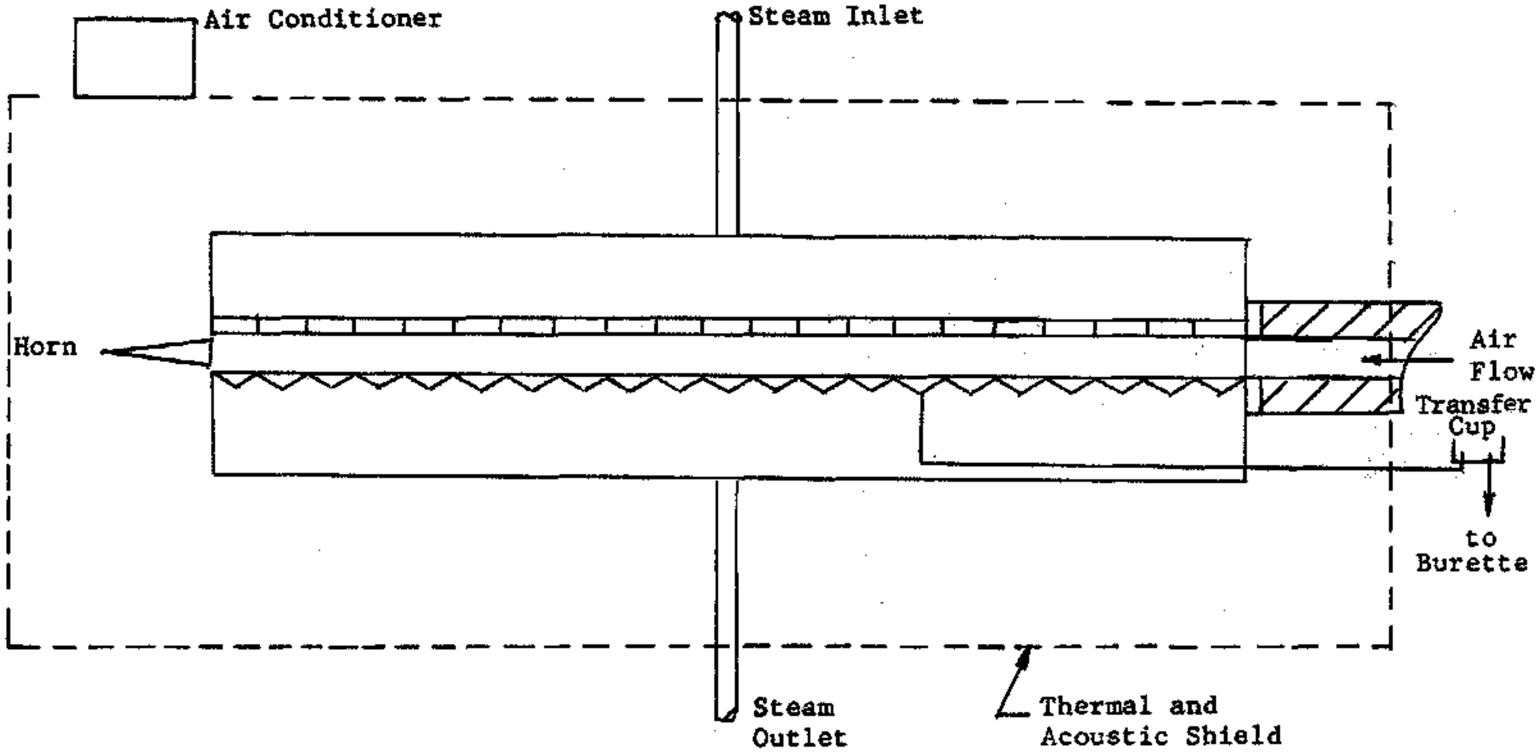


Figure 6. Schematic Diagram of Heat Transfer Section

spaced intervals. The condensate collected was transferred to burettes outside the thermal and acoustic shield via transfer cups. A total of 21 condensate collectors were fastened to the bottom of the copper tube, but only 20 were used in this investigation. The last collector was omitted since the horn extended into the exit of the heat transfer section and caused the flow to be accelerated at this point. This caused the convective heat transfer coefficient to be higher than usual.

A by-pass valve was used on the steam line to help stabilize the steam pressure within the annular chest.

b. Thermocouples. Copper-constantan (24 gauge) thermocouples were placed in the heat transfer section to measure the steam temperature and the wall temperature of the copper tube. A Leeds & Northrup 8686 Millivolt Potentiometer was used to measure the emf produced by the thermocouples.

3. Sound Generating Equipment. Figure 7 is a schematic diagram of the sound generating equipment. A Hewlett Packard Model 206A audio signal generator was connected to a Bogen Model CHA-75, 75-watt amplifier, and the amplifier was connected to a 50 watt PA-HF University driver which produced the sound field in the tube. A Model 2501 Sorensen voltage regulator supplied 115-volt, 60 cycle power to all instruments, and a Model 102 volt-amp-watt meter manufactured by The John Fluke Company measured the input power to the drivers.

The horn connected to the driver had a conical configuration and was 18 inches long with an exit diameter of two inches. Figure 8 shows the sound pressure level in the heated section for a resonant frequency of 227 cps.

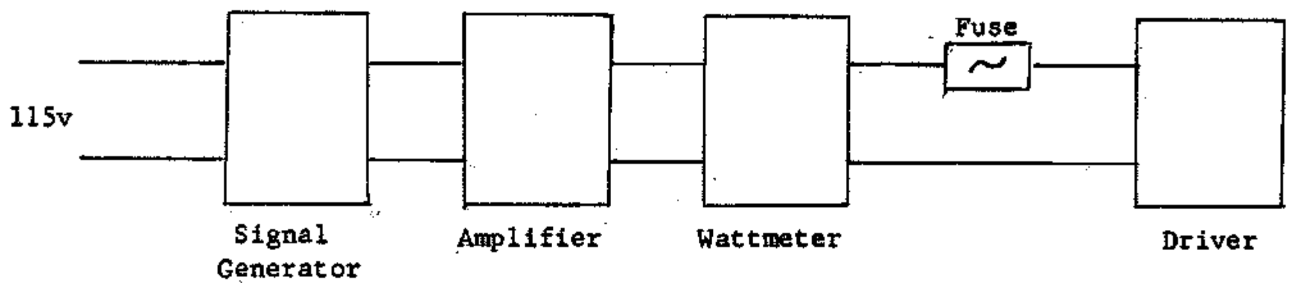


Figure 7. Schematic Diagram of Sound Generating Equipment

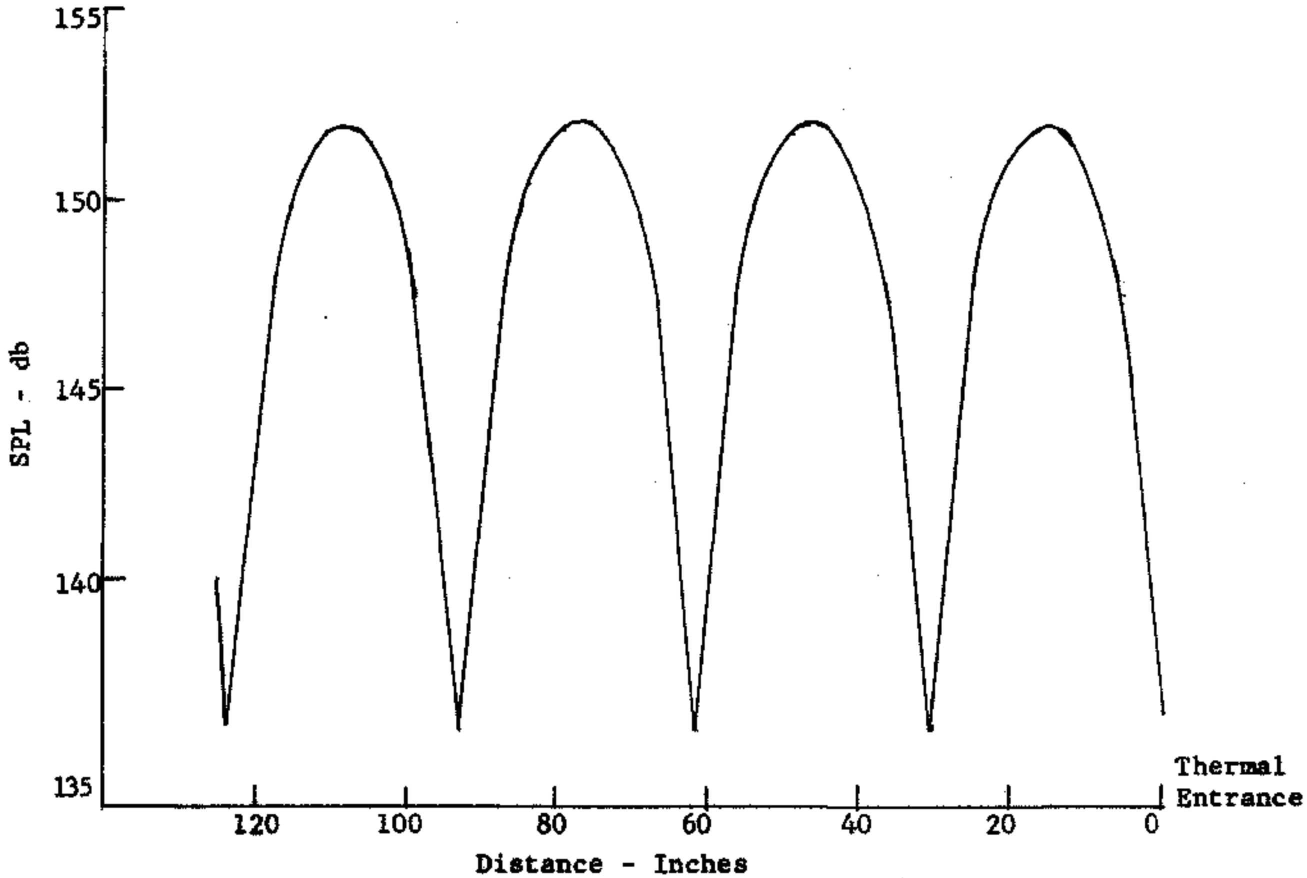


**Figure 8. Sound Pressure Level in Heated Section**

4. Sound Measuring Equipment. The sound pressure level in the tube was determined as a function of axial position by measuring the sound pressure level with a Model 1551-P1H General Radio high level microphone assembly in conjunction with a General Radio Model 1551-A SPL meter which was accurate to $\pm$ 1db (Reference 0.0002 $\mu$ bar). The microphone was mounted on one end of a 40-foot long, three-quarter inch diameter conduit so it could be inserted in the tube.

5. Thermal and Acoustic Shield. The thermal and acoustic shield was an 8 by 20 foot room eight feet high enclosing the heated section. An air conditioner held the temperature within the room constant and the walls dampened the sound so that the sound pressure level in the laboratory was at a safe level.

## Operation

Using Figure 4 as a reference, the air executed a closed cycle as follows: From the plenum chamber air entered the 3.86 inch I.D., 28.26 foot long hydrodynamic entrance section and traveled to the heat transfer section. In this entrance section which was approximately 85 diameters long, the flow became fully developed before it entered the heat transfer section. After passing through the heat transfer section the air was discharged into the air-conditioned house surrounding it, and from here the air returned to the blower compartment via the return duct. The blower discharged the air into the mixing compartment where it was filtered and then passed through the laminar flow meter. After leaving the laminar flow meter the air passed over a heating element and through a screen-bed to mix the air before it entered the plenum chamber. This completed the closed cycle.

# CHAPTER III

## EXPERIMENTAL PROCEDURE

The experiments were conducted in three phases:

1. Preliminary tests.

2. Tests without a sound field.

3. Tests with a sound field.

The following is a description of the three phases and the purpose of each:

1. Preliminary Tests. This part of the investigation was performed to determine the resonant frequencies of the tube and to determine the nodal positions of the standing sound wave in the tube. These tests were performed in the following manner: The throttle on the blower was adjusted to give the flow rate that corresponded to the desired Reynolds number and the following parameters were allowed to reach equilibrium: temperature within the thermal shield, mixing chamber temperature, plenum chamber temperature, and steam pressure within the steam chest.

The sound field was then impressed on the air flow and the nodal positions of the sound wave were determined with the microphone probe. The length of the hydrodynamic entrance section was shortened during these tests until a pressure node (velocity loop) of the sound wave was standing at the thermal entrance of the tube.

It was found during these tests that higher sound pressure levels were obtainable with this apparatus when the resonant frequency was relatively low. Thus, to reduce power requirements for the driver and to obtain as high a sound pressure level as possible, a resonant frequency of 227 cps was chosen. This frequency resulted in a wave length of 62-3/4 inches and coincided with the fifteenth harmonic for the open tube. It was also desirable to have a half wave length considerably greater than six inches since the steam condensate collectors were six inches long. This particular resonant condition also satisfied this requirement.

Other significant facts determined in this portion of the investigation were that air flow rate and inlet air temperature did not have a measurable effect on the position of the standing sound wave in this range of Reynolds numbers but air temperature did significantly affect the resonant frequency of the tube. The apparatus was capable of holding the inlet air temperature to within $\pm$ 0.25°F of the desired value, however, and this enabled the resonant frequency to be held at a constant value.

2. Tests Without a Sound Field. These tests were conducted to ascertain the reliability of the data taken using this apparatus. The results of this portion of the investigation were compared to results obtained by Latzko. Results obtained by Boelter, Young, and Iverson (8) using a similar apparatus were similar to Latzko's, and the constant, 1.4, in the empirical correlation equations is attributed to these authors. The results of these comparisons will be presented in Chapter V.

These tests were conducted as follows: the throttle was adjusted to give the flow rate that corresponded to the desired Reynolds number and the temperature within the thermal shield, mixing chamber temperature, plenum chamber temperature, and steam pressure within the steam chest were again allowed to reach equilibrium. To insure a representative value of these parameters over the duration of a test run, their values were read at approximately half-hour intervals and averaged over the length of the test run. The following readings were also taken at half hour intervals and averaged over the duration of the test run: room temperature, barometric pressure, steam temperature, tube wall temperature, and static and total pressure at the laminar flow meter.

After equilibrium had been established, the condensate levels in the burettes were noted and a stop watch was started. The time required to collect 50 ml. of condensate for each burette was recorded and from this the condensate flow rate was determined for each collection chamber. The average length of a test run was between one and two hours.

3. Tests With a Sound Field. These tests were conducted exactly the same as the tests without a sound field except that a resonant acoustic field was impressed on the air flow. Resonance of the apparatus was determined by varying the frequency of the acoustic field and measuring the sound pressure level within the tube with the microphone probe. The frequency at which the sound pressure level reached a maximum was the resonant frequency. The effect of the sound field on heat transfer was determined by comparing the test runs with and without sound at the same Reynolds number.

The resonant acoustic field had a frequency of 227 cps. Sound pressure levels from 146 db to 154 db were investigated.

# CHAPTER IV

## CALCULATIONS

In order to calculate the average heat transfer coefficient for comparison with the average heat transfer coefficient found using Latzko's equations, the following calculations were made. Calculations to obtain the Nusselt, Graetz, and Grashof numbers were also made.

Figure 9 is a schematic drawing showing the heated section and condensate collectors. This figure will be used as a reference for the calculations for chamber i ( i = 1, 2, . . . , 21).

The heat transferred from the condensing steam to the air flowing through the tube was determined by

$$\dot{q}_i = \dot{m}_i h_{fg} \tag{4-1}$$

where $\dot{q}_i$ = rate of heat transfer (Btu/min).

$\dot{m}_i$ = mass rate of flow of steam condensate ($lb_m$/min).

$h_{fg}$ = latent heat of vaporization of steam (Btu/$lb_m$).

The value of $\dot{q}_i$ found in equation (4-1) was used to determine the bulk temperature change of the air in chamber i respectively by using the reduced energy equation:

$$\Delta T_i = \frac{\dot{q}_i}{\dot{m} c_p} \tag{4-2}$$

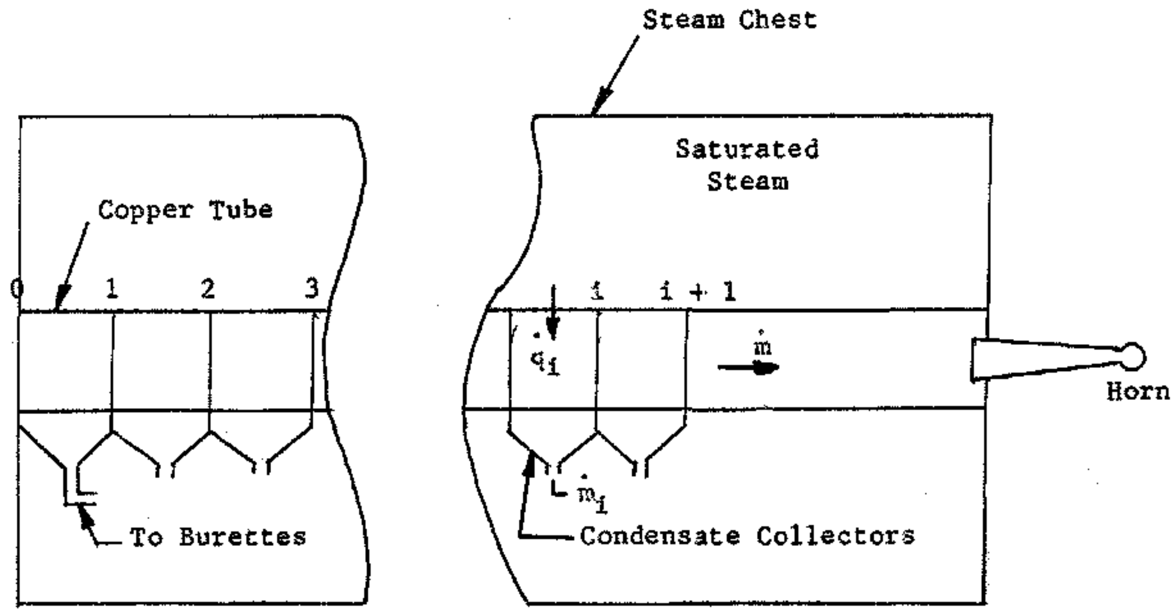


Figure 9. Schematic Diagram of Heated Tube

where $\Delta T_i$ = temperature change of the air in chamber i ($^{o}F$).

$\dot{m}$ = mass rate of flow of air ($lb_m/min$).

$C_p$ = constant pressure specific heat of air ($Btu/lb_m$-$^{o}F$).

Using the values for the changes in air temperature calculated for each chamber, the local heat transfer coefficient for each chamber was determined by

$$h_{x_i} = \frac{\dot{q}_i}{A_{x_i} \Delta T_{m_i}} \tag{4-3}$$

where $\dot{q}_i$ = rate of heat transfer calculated in equation (4-1) (Btu/min).

$A_{x_i}$ = surface area of tube wall for chamber i ($ft^2$)

$\Delta T_{m_i} = \dfrac{2T_{WALL} - T_i - T_{i-1}}{2}$ ($^{o}F$) (See Figure 10).

The local Nusselt number was calculated by multiplying the local heat transfer coefficient by $D/k_f$, i.e.

$$Nu_{x_i} = \frac{h_{x_i} D}{k_f} \quad (i = 1, 2, 3, \cdots 21) \tag{4-6}$$

where $Nu_{x_i}$ = local Nusselt number for chamber i (dimensionless).

$h_{x_i}$ = local heat transfer coefficient for chamber i (Btu/min-$ft^{2o}F$).

$D$ = diameter of tube (ft).

$k_f$ = thermal conductivity of air based on film temperature (Btu/min-ft-$^{o}F$).

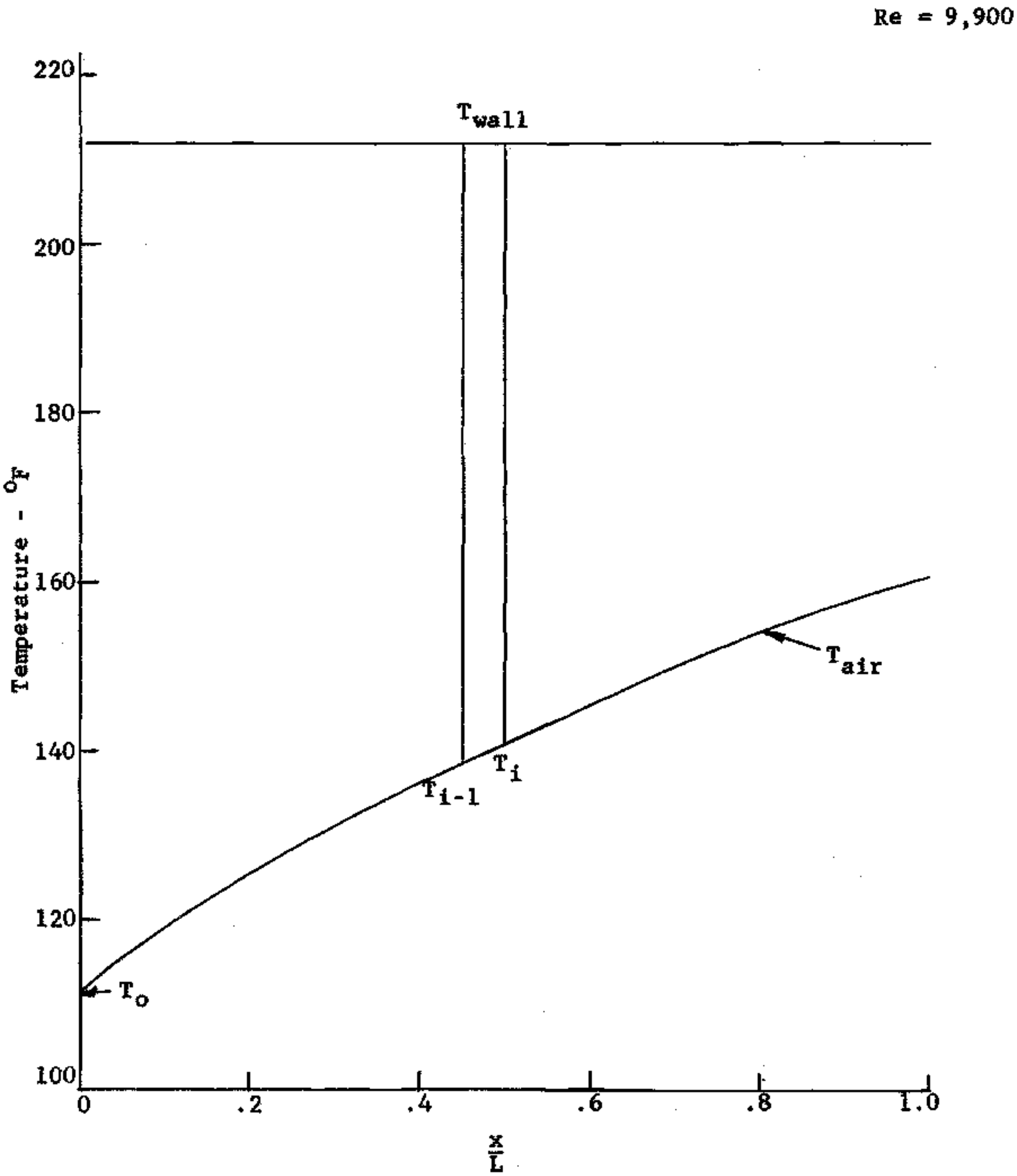


Figure 10. Typical Temperature Distribution of Air in Heated Tube

The log-mean Nusselt number was calculated in an identical manner, replacing $h_{x_i}$ with $h_{lm_i}$ in equation (4-6).

The temperature at the end of chamber i was calculated by

$$T_i = T_o + \Sigma \Delta T_i \qquad (i = 1, 2, 3, \cdots 21) \tag{4-4}$$

where $T_i$ = temperature at the end of chamber i ($^{o}$F).

$T_o$ = inlet plenum air temperature ($^{o}$F).

$\Sigma \Delta T_i$ = summation of temperature change up to section i ($^{o}$F).

These local temperatures were used with the inlet air and tube wall temperatures to calculate the log-mean temperature difference. The log-mean temperature difference was then used to calculate the average heat transfer coefficient as follows:

$$h_{lm_i} = \frac{\Sigma \dot{q}_i}{\Sigma A_{x_i} \, \Delta T_{lm_i}} \qquad (i = 1, 2, 3 \cdots 21) \tag{4-5}$$

where $h_{lm_i}$ = log-mean average heat transfer coefficient (Btu/min-ft$^{2}$$^{o}$F).

$\Sigma \dot{q}_i$ = summation of heat transferred up to section i (Btu/min).

$\Sigma A_{x_i}$ = surface area of tube wall up to section i (ft$^2$).

$\Delta T_i$ = log-mean temperature ($^{o}$F).

$$= \frac{T_i - T_o}{\ln\left(\frac{T_{WALL} - T_o}{T_{WALL} - T_i}\right)}$$

The Reynolds number was based on the properties at the inlet plenum chamber and the tube diameter.

The experimental data taken and used for the above calculations were steam condensate flow rate for each chamber, air density, air flow rate, inlet air temperature, and tube wall temperature. Property data for steam were taken from reference (6) and property data for air were taken from reference (7).

# CHAPTER V

## CORRELATION OF NO-SOUND DATA

Latzko's empirical equations for fully developed turbulent flow were programmed for an electronic digital computer and were used as the standard for comparison of all no-sound data.

From his experimental data, Latzko developed the following two equations for the asymptotic and average heat transfer coefficients for fully developed turbulent flow:

$$h_{\infty} = \frac{(0.23)(c_p)(G)}{(DG/\mu_f)^{0.20}(\mu_f c_p / k_f)^{0.67}} \qquad (5\text{-}1)$$

$$\frac{h_m}{h_{\infty}} = 1 + \frac{1.4\,D}{L} \quad \left(\frac{L}{D} > 5\right) \qquad (5\text{-}2)$$

The asymptotic heat transfer coefficient, $h_{\infty}$, is the local heat transfer coefficient at a position in the flow where the temperature distribution is fully developed.

The computer program was written to calculate the asymptotic and average heat transfer coefficients for chambers 6 through 20 (6.3 L/D 29.6). Figure 11 shows the comparison between the experimental average heat transfer coefficient and the one computed from equations (5-1) and (5-2) and for $9,900 < Re < 46,500$.

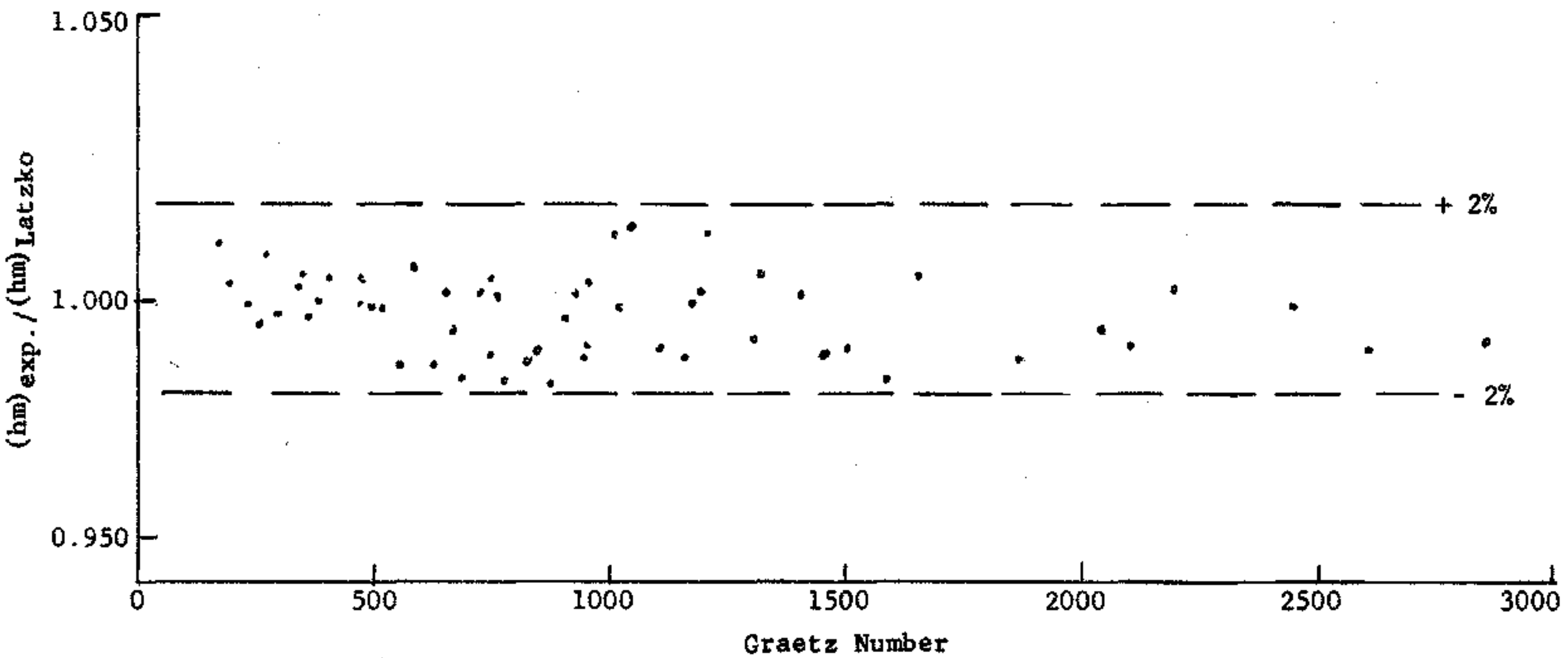


Figure 11. Correlation of No-Sound Data

# CHAPTER VI

## DISCUSSION

Figure 11 shows that the average heat transfer coefficients for the no-sound runs found in this investigation were within $\pm$ 2 per cent of the values predicted using Latzko's empirical equations. The small variation in the average heat transfer coefficient from the value predicted indicates that the apparatus is quite reliable, and that the correlation with Latzko's equations is excellent.

Figure 12 shows a typical correlation of the experimental no-sound average heat transfer coefficients with the predicted values. Correlations for Reynolds numbers other than the one shown were very similar and are not shown. It should be noted that all correlated data were for chambers 6 through 20. Chamber 21 was omitted because the horn extended into the exit of the heated tube and accelerated the air flow at this point. This caused a significant increase in the heat transfer coefficient at this point. Chambers 1 through 5 were omitted because Latzko's equations are not valid for $L/D < 5$. Chamber 6 had a L/D value of 6.3, and this chamber was used as the first one for correlating data with Latzko's equations.

The shaded portion of Figure 13 shows the range of Reynolds numbers and sound pressure levels investigated in this work. The critical sound pressure levels for developing flow as determined by Eastwood (2) are superimposed.

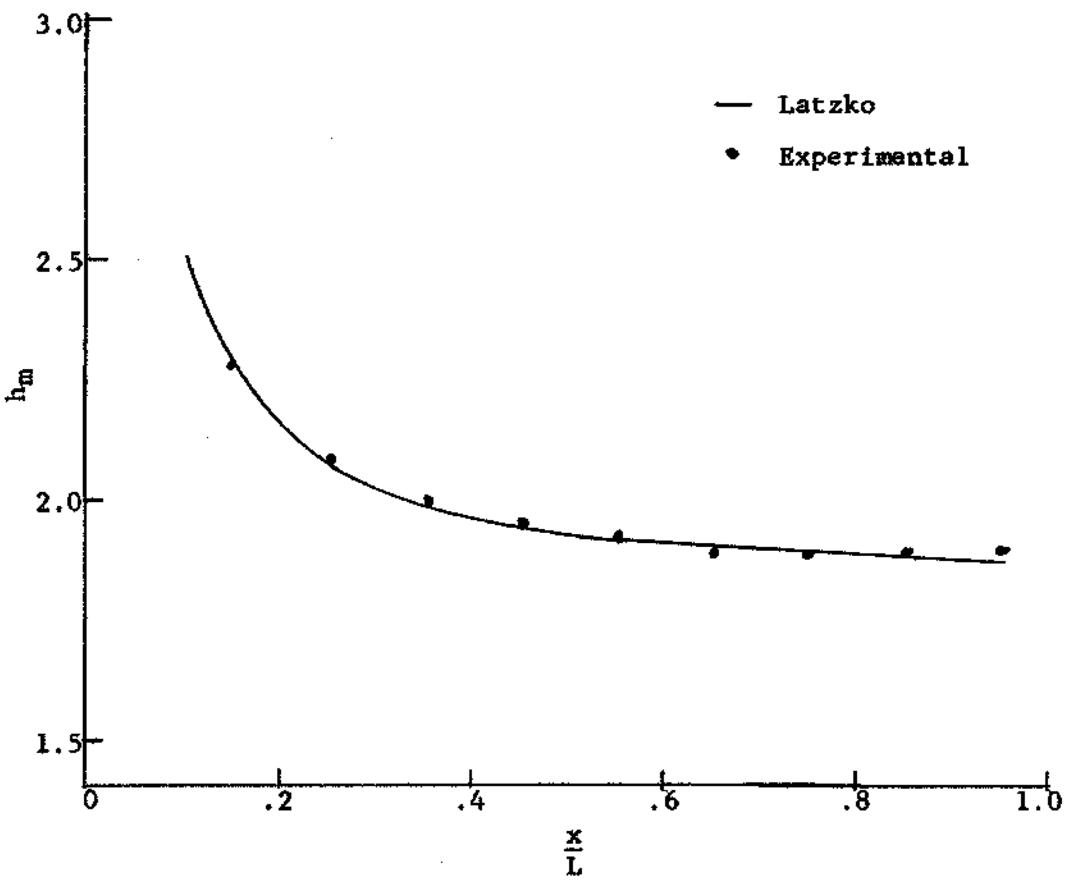


Figure 12. Heat Transfer Coefficient vs. $\frac{x}{L}$ Re = 9,900

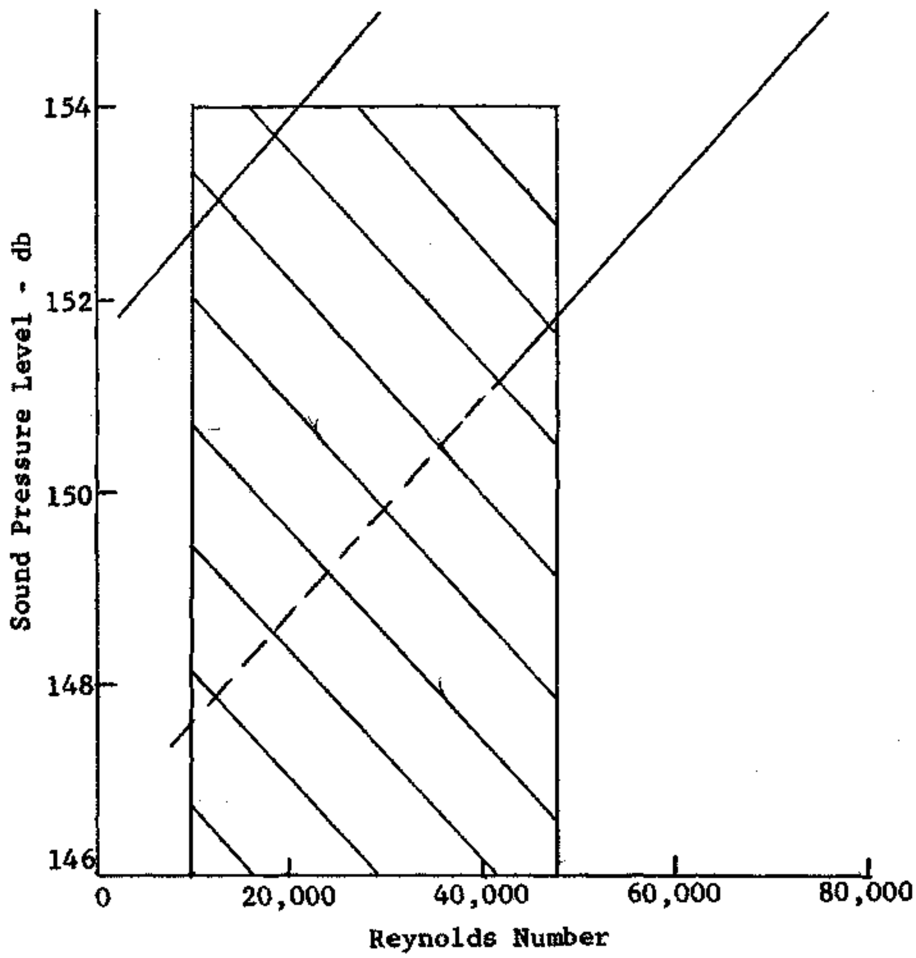


Figure 13. Region of Investigation

Except for one run at a Reynolds number of 9,900 and SPL of 154 db, no effect on heat transfer by the sound field could be determined in the shaded portion of Figure 13. This indicates that if the threshold SPL for fully developed flow exists, it is not identical to that determined by Eastwood (1) for developing flow. It also indicates that the line for threshold SPL beginning at approximately Re = 40,000 cannot be extended to lower Reynolds numbers.

Figure 14 (which is the same as Figure 2) shows typical results obtained by Eastwood in this region. This phenomena could not be reproduced for fully developed flow at the obtainable sound pressure levels.

A measurable effect on heat transfer by the sound field was obtained at a Reynolds number of 9,900 and a sound pressure level of 154 db, however. Figure 15 shows this effect and it is seen to be similar to that obtained by Eastwood in this range of Reynolds numbers. This phenomenon, which is caused by standing vortices that form secondary flows, is described by Purdy (4).

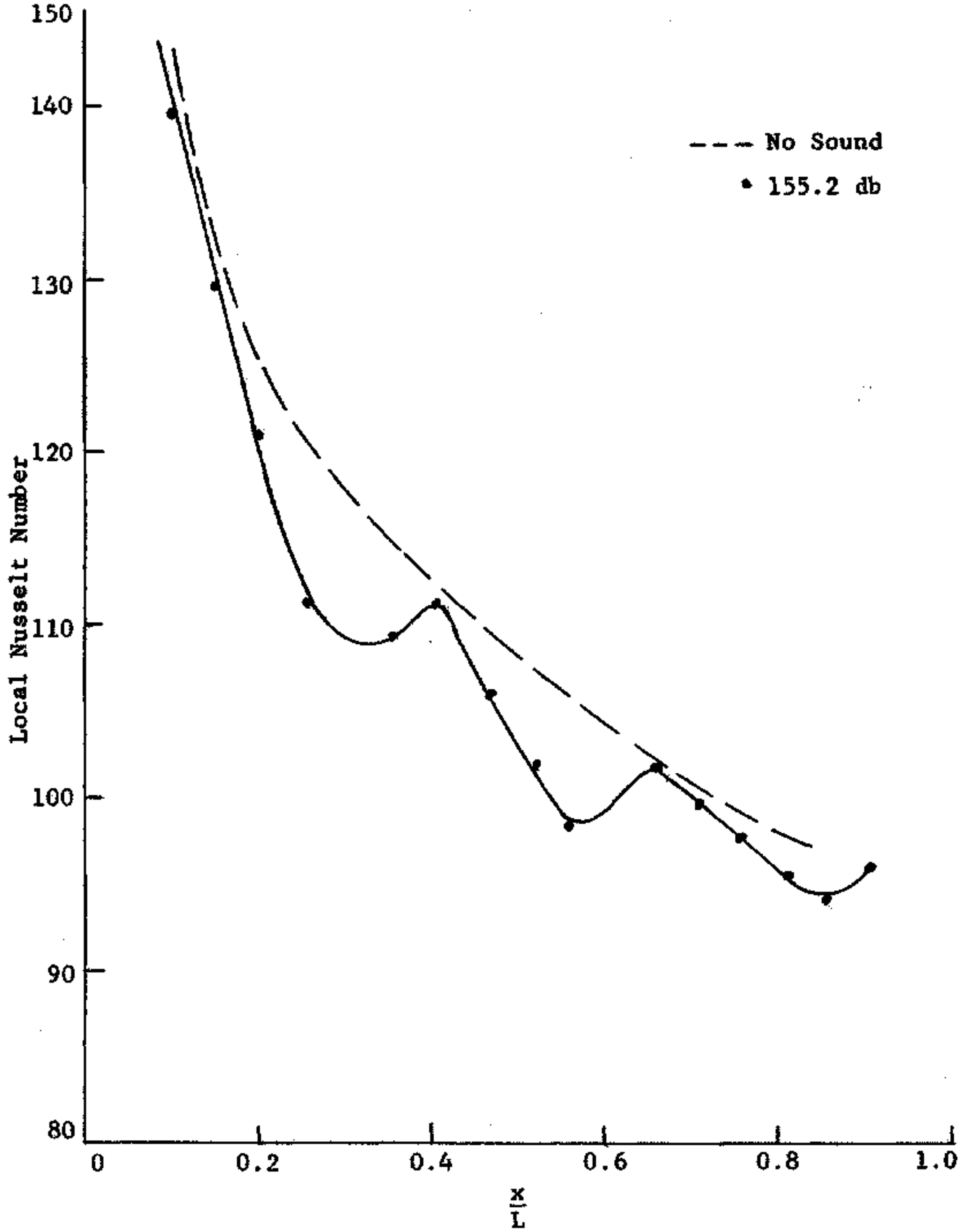


Figure 14. Local Nusselt Number vs. $\frac{x}{L}$ Re = 43,000

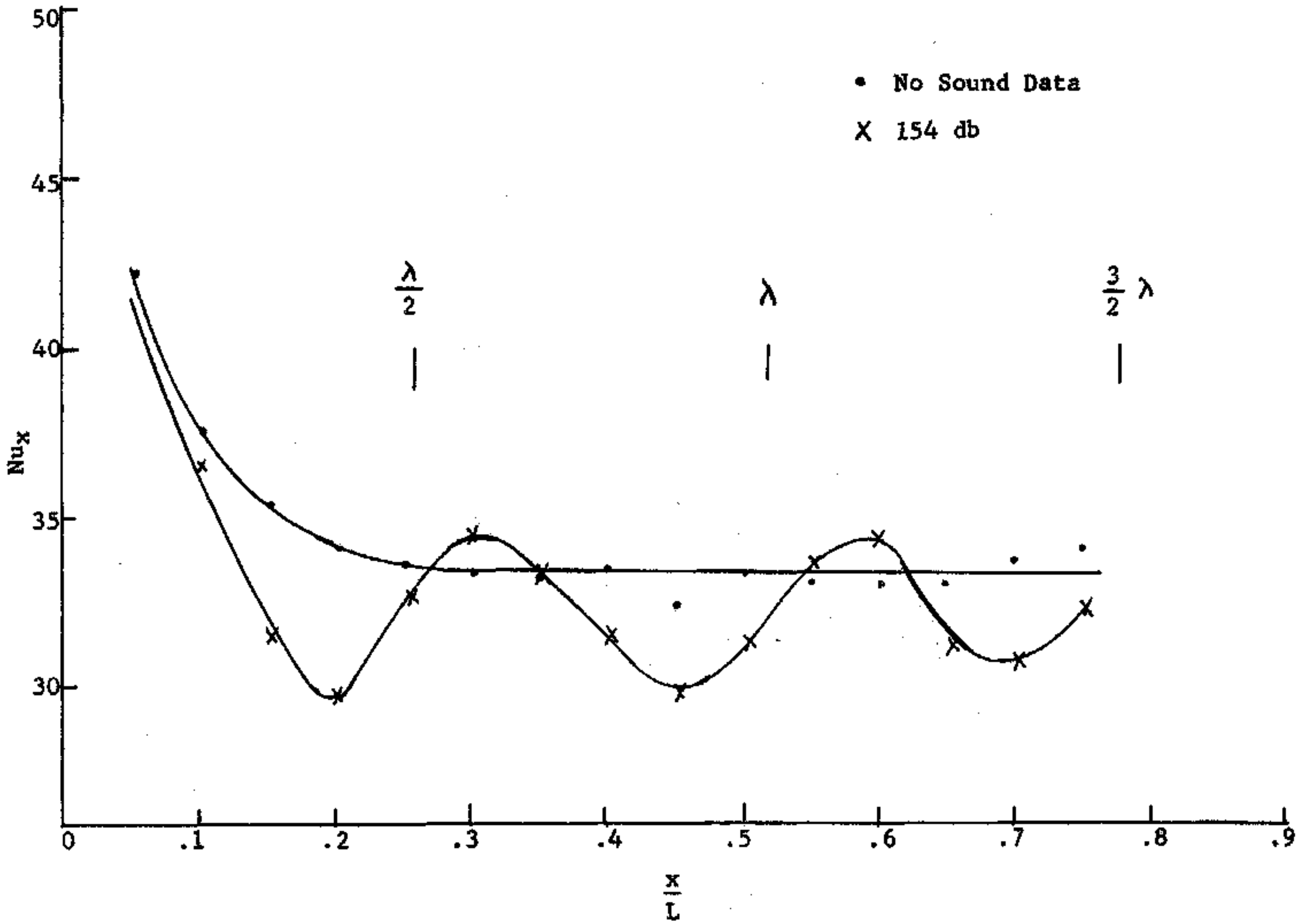


Figure 15. Nusselt Number vs. $\frac{x}{L}$ Re = 9,900

# CHAPTER VII

## CONCLUSIONS AND RECOMMENDATIONS

The resonant acoustic field did not produce any measurable effects on heat transfer between Reynolds numbers of 15,800 and 46,500 with sound pressure levels up to 154 db. At a Reynolds number of 9,900 and a sound pressure level of 154 db the resonant acoustic field produced a significant change in the heat transfer characteristics. However, it was not possible to determine the threshold sound pressure levels for this flow because insufficient sound intensity was available to obtain the necessary data.

The excellent correlation of the no-sound data with Latzko's results for fully developed turbulent flow indicates that the system is quite reliable and furnishes excellent heat transfer data.

It is recommended that a new sound system that will generate considerably higher sound pressure levels be obtained and used for further investigations.

# APPENDICES

# APPENDIX A

## SUMMARY TABULATION OF LOCAL AND AVERAGE DATA

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 101 Reynolds Number = 9910.4 Max. SPL = .0 DB
$T_o$ = 110.52 F $T_s$ = 211.94 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 114.071 | 3.59482 | 67.3007 | 17683.4 |
| 2 | 115.991 | 2.25866 | 42.2055 | 4053.58 |
| 3 | 119.334 | 2.01786 | 37.6372 | 2104.25 |
| 4 | 122.416 | 1.90179 | 35.3934 | 1278.62 |
| 5 | 125.263 | 1.83891 | 34.1533 | 917.379 |
| 6 | 127.988 | 1.81715 | 33.6846 | 715.715 |
| 7 | 130.628 | 1.80899 | 33.4717 | 586.101 |
| 8 | 133.188 | 1.79591 | 33.1707 | 495.243 |
| 9 | 135.639 | 1.82111 | 33.5787 | 429.442 |
| 10 | 138.026 | 1.75460 | 32.2992 | 378.433 |
| 11 | 140.340 | 1.81494 | 33.3566 | 335.070 |
| 12 | 142.589 | 1.80820 | 33.1814 | 305.657 |
| 13 | 144.779 | 1.80882 | 33.1431 | 278.767 |
| 14 | 146.896 | 1.80580 | 33.0396 | 256.123 |
| 15 | 148.994 | 1.84660 | 33.7383 | 236.831 |
| 16 | 151.047 | 1.88414 | 34.3761 | 220.261 |
| 17 | 153.058 | 1.90142 | 34.6440 | 205.860 |
| 18 | 155.042 | 1.92760 | 35.0741 | 193.145 |
| 19 | 157.000 | 1.97036 | 35.8049 | 181.852 |
| 20 | 158.931 | 2.01467 | 36.5627 | 171.789 |
| 21 | 160.781 | 2.00751 | 36.3868 | 162.784 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 114.071 | 3.59521 | 67.3079 | 5937.09 |
| 2 | 115.991 | 2.93801 | 54.9673 | 3115.17 |
| 3 | 119.334 | 2.48134 | 46.3696 | 1602.61 |
| 4 | 122.416 | 2.28372 | 42.6310 | 1073.63 |
| 5 | 125.263 | 2.16973 | 40.4633 | 809.351 |
| 6 | 127.988 | 2.09643 | 39.0594 | 649.254 |
| 7 | 130.628 | 2.04566 | 38.0789 | 541.433 |
| 8 | 133.188 | 2.00728 | 37.3314 | 463.662 |
| 9 | 135.639 | 1.98205 | 36.8311 | 406.617 |
| 10 | 138.026 | 1.95424 | 36.2844 | 360.347 |
| 11 | 140.340 | 1.93857 | 35.9650 | 324.536 |
| 12 | 142.589 | 1.92505 | 35.6866 | 294.983 |
| 13 | 144.779 | 1.91378 | 35.4510 | 270.221 |
| 14 | 146.896 | 1.90398 | 35.2439 | 249.272 |
| 15 | 148.994 | 1.89846 | 35.1165 | 231.302 |
| 16 | 151.047 | 1.89619 | 35.0498 | 215.852 |
| 17 | 153.058 | 1.89525 | 35.0085 | 202.279 |
| 18 | 155.042 | 1.89594 | 34.9974 | 190.229 |
| 19 | 157.000 | 1.89890 | 35.0287 | 179.522 |
| 20 | 158.931 | 1.90388 | 35.0974 | 169.953 |
| 21 | 160.781 | 1.90797 | 35.1506 | 161.374 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 102 Reynolds Number = 15808.6 Max. SPL = .0 DB
$T_o$ = 109.87 F $T_s$ = 212.00 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 112.772 | 4.62874 | 86.7120 | 28179.1 |
| 2 | 114.398 | 2.99941 | 56.1008 | 6461.61 |
| 3 | 117.294 | 2.72979 | 50.9777 | 3355.13 |
| 4 | 119.961 | 2.55492 | 47.6202 | 2039.30 |
| 5 | 122.508 | 2.54312 | 47.3150 | 1463.50 |
| 6 | 124.888 | 2.44285 | 45.3724 | 1142.03 |
| 7 | 127.182 | 2.40520 | 44.6014 | 935.433 |
| 8 | 129.419 | 2.38957 | 44.2428 | 790.599 |
| 9 | 131.584 | 2.43592 | 45.0330 | 685.693 |
| 10 | 133.697 | 2.34118 | 43.2184 | 604.359 |
| 11 | 135.743 | 2.40533 | 44.3396 | 535.204 |
| 12 | 137.795 | 2.46137 | 45.3094 | 488.297 |
| 13 | 139.768 | 2.41915 | 44.4715 | 445.399 |
| 14 | 141.703 | 2.43837 | 44.7654 | 409.272 |
| 15 | 143.572 | 2.41662 | 44.3091 | 378.495 |
| 16 | 145.409 | 2.46218 | 45.0880 | 352.064 |
| 17 | 147.225 | 2.49211 | 45.5798 | 329.088 |
| 18 | 148.982 | 2.46353 | 45.0029 | 308.803 |
| 19 | 150.710 | 2.49074 | 45.4467 | 290.790 |
| 20 | 152.441 | 2.56925 | 46.8248 | 274.737 |
| 21 | 154.095 | 2.53396 | 46.1293 | 260.367 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 112.772 | 4.62907 | 86.7181 | 9460.93 |
| 2 | 114.398 | 3.82528 | 71.6199 | 4964.62 |
| 3 | 117.294 | 3.27986 | 61.3462 | 2554.46 |
| 4 | 119.961 | 3.03201 | 56.6581 | 1711.54 |
| 5 | 122.508 | 2.90535 | 54.2432 | 1290.37 |
| 6 | 124.888 | 2.80905 | 52.4022 | 1035.24 |
| 7 | 127.182 | 2.73797 | 51.0356 | 863.425 |
| 8 | 129.419 | 2.68458 | 50.0018 | 739.486 |
| 9 | 131.584 | 2.65088 | 49.3372 | 648.568 |
| 10 | 133.697 | 2.61306 | 48.5978 | 574.819 |
| 11 | 135.743 | 2.59001 | 48.1352 | 517.742 |
| 12 | 137.795 | 2.57601 | 47.8413 | 470.625 |
| 13 | 139.768 | 2.56082 | 47.5268 | 431.151 |
| 14 | 141.703 | 2.54938 | 47.2830 | 397.750 |
| 15 | 143.572 | 2.53803 | 47.0424 | 369.104 |
| 16 | 145.409 | 2.53127 | 46.8875 | 344.474 |
| 17 | 147.225 | 2.52718 | 46.7826 | 322.834 |
| 18 | 148.982 | 2.52186 | 46.6561 | 303.626 |
| 19 | 150.710 | 2.51863 | 46.5688 | 286.559 |
| 20 | 152.441 | 2.51987 | 46.5641 | 271.302 |
| 21 | 154.095 | 2.51920 | 46.5254 | 257.624 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 103 Reynolds Number = 20084.1 Max. SPL = .0 DB
$T_o$ = 110.57 F $T_s$ = 211.75 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 113.230 | 5.44782 | 102.032 | 35859.2 |
| 2 | 114.804 | 3.72086 | 69.5859 | 8223.54 |
| 3 | 117.563 | 3.32942 | 62.1718 | 4270.28 |
| 4 | 120.127 | 3.14374 | 58.5964 | 2595.76 |
| 5 | 122.547 | 3.08703 | 57.4404 | 1862.99 |
| 6 | 124.822 | 2.97890 | 55.3389 | 1453.89 |
| 7 | 127.037 | 2.96151 | 54.9309 | 1190.94 |
| 8 | 129.223 | 2.97589 | 55.1143 | 1006.59 |
| 9 | 131.311 | 2.99083 | 55.3100 | 873.065 |
| 10 | 133.368 | 2.89957 | 53.5464 | 769.540 |
| 11 | 135.348 | 2.95743 | 54.5398 | 681.512 |
| 12 | 137.259 | 2.91040 | 53.6013 | 621.826 |
| 13 | 139.153 | 2.94155 | 54.1050 | 567.239 |
| 14 | 140.975 | 2.90515 | 53.3683 | 521.263 |
| 15 | 142.784 | 2.95667 | 54.2480 | 482.091 |
| 16 | 144.565 | 3.01179 | 55.1921 | 448.443 |
| 17 | 146.313 | 3.02620 | 55.3903 | 419.194 |
| 18 | 148.011 | 2.99673 | 54.7871 | 393.372 |
| 19 | 149.695 | 3.05343 | 55.7602 | 370.438 |
| 20 | 151.352 | 3.08906 | 56.3476 | 350.001 |
| 21 | 152.957 | 3.08504 | 56.2127 | 331.709 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 113.230 | 5.44814 | 102.038 | 12039.4 |
| 2 | 114.804 | 4.59081 | 85.9347 | 6317.83 |
| 3 | 117.563 | 3.96181 | 74.0895 | 3250.88 |
| 4 | 120.127 | 3.68120 | 68.7806 | 2178.24 |
| 5 | 122.547 | 3.52727 | 65.8491 | 1642.29 |
| 6 | 124.822 | 3.41292 | 63.6642 | 1317.63 |
| 7 | 127.037 | 3.33297 | 62.1252 | 1098.98 |
| 8 | 129.223 | 3.27745 | 61.0443 | 941.244 |
| 9 | 131.311 | 3.23836 | 60.2726 | 825.543 |
| 10 | 133.368 | 3.19657 | 59.4527 | 731.684 |
| 11 | 135.348 | 3.16984 | 58.9154 | 659.046 |
| 12 | 137.259 | 3.14359 | 58.3890 | 599.099 |
| 13 | 139.153 | 3.12417 | 57.9905 | 548.864 |
| 14 | 140.975 | 3.10492 | 57.5971 | 506.363 |
| 15 | 142.784 | 3.09205 | 57.3227 | 469.905 |
| 16 | 144.565 | 3.08461 | 57.1498 | 438.557 |
| 17 | 146.313 | 3.07896 | 57.0110 | 411.016 |
| 18 | 148.011 | 3.07221 | 56.8529 | 386.569 |
| 19 | 149.695 | 3.06933 | 56.7669 | 364.845 |
| 20 | 151.352 | 3.06864 | 56.7220 | 345.429 |
| 21 | 152.957 | 3.06779 | 56.6752 | 328.019 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 104 Reynolds Number = 25239.3 Max. SPL = .0 DB
$T_o$ = 110.87 F $T_s$ = 211.81 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 113.291 | 6.23146 | 116.690 | 45080.1 |
| 2 | 114.714 | 4.22639 | 79.0378 | 10339.5 |
| 3 | 117.289 | 3.89881 | 72.8107 | 5369.71 |
| 4 | 119.754 | 3.78299 | 70.5244 | 3264.38 |
| 5 | 122.021 | 3.61649 | 67.3105 | 2343.07 |
| 6 | 124.156 | 3.49064 | 64.8696 | 1828.72 |
| 7 | 126.249 | 3.48705 | 64.7086 | 1498.12 |
| 8 | 128.279 | 3.43621 | 63.6749 | 1266.34 |
| 9 | 130.264 | 3.52880 | 65.3008 | 1098.45 |
| 10 | 132.179 | 3.34421 | 61.8025 | 968.284 |
| 11 | 134.040 | 3.43854 | 63.4639 | 857.596 |
| 12 | 135.835 | 3.37215 | 62.1609 | 782.551 |
| 13 | 137.615 | 3.40665 | 62.7205 | 713.909 |
| 14 | 139.330 | 3.36059 | 61.7991 | 656.093 |
| 15 | 141.031 | 3.40851 | 62.6077 | 606.833 |
| 16 | 142.683 | 3.42096 | 62.7652 | 564.522 |
| 17 | 144.329 | 3.47767 | 63.7345 | 527.742 |
| 18 | 145.945 | 3.47559 | 63.6263 | 495.263 |
| 19 | 147.537 | 3.50732 | 64.1377 | 466.416 |
| 20 | 149.110 | 3.55642 | 64.9663 | 440.708 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 113.291 | 6.23178 | 116.696 | 15135.3 |
| 2 | 114.714 | 5.23734 | 98.0260 | 7942.83 |
| 3 | 117.289 | 4.56795 | 85.4207 | 4087.30 |
| 4 | 119.754 | 4.29597 | 80.2659 | 2738.77 |
| 5 | 122.021 | 4.11970 | 76.9120 | 2065.02 |
| 6 | 124.156 | 3.98836 | 74.4050 | 1656.87 |
| 7 | 126.249 | 3.89921 | 72.6892 | 1381.98 |
| 8 | 128.279 | 3.82787 | 71.3091 | 1183.69 |
| 9 | 130.264 | 3.78652 | 70.4905 | 1038.22 |
| 10 | 132.179 | 3.73250 | 69.4389 | 920.228 |
| 11 | 134.040 | 3.69984 | 68.7873 | 828.906 |
| 12 | 135.835 | 3.66698 | 68.1341 | 753.539 |
| 13 | 137.615 | 3.64229 | 67.6339 | 690.380 |
| 14 | 139.330 | 3.61784 | 67.1403 | 636.945 |
| 15 | 141.031 | 3.60025 | 66.7749 | 591.106 |
| 16 | 142.683 | 3.58597 | 66.4723 | 551.697 |
| 17 | 144.329 | 3.57692 | 66.2671 | 517.069 |
| 18 | 145.945 | 3.56874 | 66.0788 | 486.327 |
| 19 | 147.537 | 3.56321 | 65.9405 | 459.012 |
| 20 | 149.110 | 3.56086 | 65.8616 | 434.597 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 105 Reynolds Number = 31905.1 Max. SPL = .0 DB
$T_o$ = 110.35 F $T_s$ = 211.89 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 112.573 | 7.18025 | 134.508 | 56937.3 |
| 2 | 113.905 | 4.95397 | 92.6887 | 13060.4 |
| 3 | 116.336 | 4.60032 | 85.9593 | 6783.28 |
| 4 | 118.579 | 4.29350 | 80.0964 | 4124.25 |
| 5 | 120.739 | 4.28873 | 79.8857 | 2960.58 |
| 6 | 122.756 | 4.09821 | 76.2269 | 2310.86 |
| 7 | 124.720 | 4.05785 | 75.3728 | 1893.26 |
| 8 | 126.666 | 4.07932 | 75.6699 | 1600.45 |
| 9 | 128.510 | 4.05346 | 75.0924 | 1388.37 |
| 10 | 130.343 | 3.95114 | 73.1046 | 1223.94 |
| 11 | 132.126 | 4.05806 | 74.9900 | 1084.08 |
| 12 | 133.861 | 4.01070 | 74.0260 | 989.264 |
| 13 | 135.568 | 4.01240 | 73.9704 | 902.530 |
| 14 | 137.230 | 3.99274 | 73.5236 | 829.473 |
| 15 | 138.848 | 3.97241 | 73.0678 | 767.228 |
| 16 | 140.445 | 4.04124 | 74.2528 | 713.766 |
| 17 | 142.008 | 4.02956 | 73.9590 | 667.292 |
| 18 | 143.563 | 4.07398 | 74.6954 | 626.255 |
| 19 | 145.078 | 4.06092 | 74.3788 | 589.803 |
| 20 | 146.572 | 4.09675 | 74.9590 | 557.323 |
| 21 | 148.033 | 4.11355 | 75.1914 | 528.247 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 112.573 | 7.18056 | 134.514 | 19116.2 |
| 2 | 113.905 | 6.07382 | 113.728 | 10032.3 |
| 3 | 116.336 | 5.33532 | 99.8163 | 5162.77 |
| 4 | 118.579 | 4.97691 | 93.0384 | 3459.68 |
| 5 | 120.739 | 4.79673 | 89.6029 | 2608.67 |
| 6 | 122.756 | 4.65041 | 86.8089 | 2093.16 |
| 7 | 124.720 | 4.54514 | 84.7863 | 1745.96 |
| 8 | 126.666 | 4.47242 | 83.3735 | 1495.48 |
| 9 | 128.510 | 4.41570 | 82.2636 | 1311.76 |
| 10 | 130.343 | 4.35843 | 81.1453 | 1162.71 |
| 11 | 132.126 | 4.32448 | 80.4638 | 1047.35 |
| 12 | 133.861 | 4.29225 | 79.8164 | 952.145 |
| 13 | 135.568 | 4.26540 | 79.2704 | 872.362 |
| 14 | 137.230 | 4.24112 | 78.7741 | 804.855 |
| 15 | 138.848 | 4.21886 | 78.3169 | 746.953 |
| 16 | 140.445 | 4.20424 | 78.0028 | 697.167 |
| 17 | 142.008 | 4.19069 | 77.7097 | 653.427 |
| 18 | 143.563 | 4.18122 | 77.4926 | 614.590 |
| 19 | 145.078 | 4.17208 | 77.2830 | 580.085 |
| 20 | 146.572 | 4.16582 | 77.1276 | 549.245 |
| 21 | 148.033 | 4.16098 | 76.9995 | 521.587 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 106 Reynolds Number = 37364.3 Max. SPL = .0 DB

$T_o$ = 110.57 F $T_s$ = 211.97 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 112.686 | 8.01975 | 150.208 | 66704.2 |
| 2 | 113.967 | 5.58152 | 104.417 | 15301.6 |
| 3 | 116.315 | 5.20232 | 97.2012 | 7947.69 |
| 4 | 118.496 | 4.88634 | 91.1543 | 4832.46 |
| 5 | 120.591 | 4.86381 | 90.5998 | 3469.12 |
| 6 | 122.562 | 4.67827 | 87.0216 | 2707.90 |
| 7 | 124.479 | 4.62621 | 85.9379 | 2218.61 |
| 8 | 126.336 | 4.54213 | 84.2666 | 1875.58 |
| 9 | 128.150 | 4.64858 | 86.1329 | 1627.11 |
| 10 | 129.930 | 4.47071 | 82.7352 | 1434.44 |
| 11 | 131.642 | 4.53510 | 83.8265 | 1270.58 |
| 12 | 133.311 | 4.48568 | 82.8174 | 1159.50 |
| 13 | 134.968 | 4.52619 | 83.4706 | 1057.88 |
| 14 | 136.566 | 4.45941 | 82.1478 | 972.291 |
| 15 | 138.159 | 4.53362 | 83.4244 | 899.356 |
| 16 | 139.709 | 4.54791 | 83.5981 | 836.708 |
| 17 | 141.243 | 4.57900 | 84.0816 | 782.248 |
| 18 | 142.755 | 4.58512 | 84.1073 | 734.158 |
| 19 | 144.228 | 4.56362 | 83.6284 | 691.445 |
| 20 | 145.699 | 4.66048 | 85.3185 | 653.382 |
| 21 | 147.123 | 4.62811 | 84.6433 | 619.307 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 112.686 | 8.02003 | 150.214 | 22395.4 |
| 2 | 113.967 | 6.80642 | 127.426 | 11753.4 |
| 3 | 116.315 | 6.00144 | 112.264 | 6048.68 |
| 4 | 118.496 | 5.61678 | 104.989 | 4053.43 |
| 5 | 120.591 | 5.41918 | 101.222 | 3056.44 |
| 6 | 122.562 | 5.26328 | 98.2431 | 2452.48 |
| 7 | 124.479 | 5.14964 | 96.0580 | 2045.71 |
| 8 | 126.336 | 5.05597 | 94.2503 | 1752.29 |
| 9 | 128.150 | 4.99985 | 93.1457 | 1537.03 |
| 10 | 129.930 | 4.93465 | 91.8746 | 1362.41 |
| 11 | 131.642 | 4.89041 | 90.9972 | 1227.27 |
| 12 | 133.311 | 4.84951 | 90.1841 | 1115.73 |
| 13 | 134.968 | 4.81858 | 89.5578 | 1022.25 |
| 14 | 136.566 | 4.78726 | 88.9267 | 943.173 |
| 15 | 138.159 | 4.76564 | 88.4765 | 875.329 |
| 16 | 139.709 | 4.74801 | 88.1023 | 816.998 |
| 17 | 141.243 | 4.73445 | 87.8044 | 765.748 |
| 18 | 142.755 | 4.72273 | 87.5415 | 720.246 |
| 19 | 144.228 | 4.71112 | 87.2823 | 679.819 |
| 20 | 145.699 | 4.70584 | 87.1406 | 643.682 |
| 21 | 147.123 | 4.69946 | 86.9799 | 611.275 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 107 Reynolds Number = 39861.5 Max. SPL = .0 DB
$T_o$ = 110.53 F $T_s$ = 211.85 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 112.596 | 8.34059 | 156.238 | 71197.2 |
| 2 | 113.866 | 5.91023 | 110.583 | 16332.6 |
| 3 | 116.193 | 5.50224 | 102.821 | 8483.31 |
| 4 | 118.340 | 5.13249 | 95.7632 | 5158.23 |
| 5 | 120.391 | 5.07956 | 94.6382 | 3703.08 |
| 6 | 122.334 | 4.91537 | 91.4529 | 2890.59 |
| 7 | 124.225 | 4.86432 | 90.3835 | 2368.34 |
| 8 | 126.075 | 4.82162 | 89.4748 | 2002.18 |
| 9 | 127.871 | 4.90356 | 90.8816 | 1736.95 |
| 10 | 129.644 | 4.74144 | 87.7694 | 1531.28 |
| 11 | 131.346 | 4.80282 | 88.7999 | 1356.37 |
| 12 | 133.010 | 4.76075 | 87.9210 | 1237.80 |
| 13 | 134.651 | 4.77224 | 88.0339 | 1129.33 |
| 14 | 136.228 | 4.68281 | 86.2895 | 1037.96 |
| 15 | 137.787 | 4.72347 | 86.9459 | 960.122 |
| 16 | 139.320 | 4.78037 | 87.9008 | 893.255 |
| 17 | 140.814 | 4.74041 | 87.0767 | 835.131 |
| 18 | 142.297 | 4.77721 | 87.6641 | 783.808 |
| 19 | 143.763 | 4.82477 | 88.4486 | 738.215 |
| 20 | 145.219 | 4.89527 | 89.6526 | 697.582 |
| 21 | 146.643 | 4.91391 | 89.9065 | 661.205 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 112.596 | 8.34092 | 156.244 | 23903.9 |
| 2 | 113.866 | 7.12770 | 133.458 | 12545.1 |
| 3 | 116.193 | 6.31082 | 118.068 | 6456.17 |
| 4 | 118.340 | 5.90445 | 110.382 | 4326.55 |
| 5 | 120.391 | 5.68863 | 106.272 | 3262.44 |
| 6 | 122.334 | 5.52585 | 103.162 | 2617.80 |
| 7 | 124.225 | 5.40773 | 100.890 | 2183.62 |
| 8 | 126.075 | 5.31670 | 99.1289 | 1870.43 |
| 9 | 127.871 | 5.25953 | 98.0022 | 1640.67 |
| 10 | 129.644 | 5.19526 | 96.7452 | 1454.28 |
| 11 | 131.346 | 5.15144 | 95.8729 | 1310.03 |
| 12 | 133.010 | 5.11153 | 95.0757 | 1190.97 |
| 13 | 134.651 | 5.07906 | 94.4182 | 1091.19 |
| 14 | 136.228 | 5.04470 | 93.7286 | 1006.78 |
| 15 | 137.787 | 5.01810 | 93.1844 | 934.377 |
| 16 | 139.320 | 4.99899 | 92.7805 | 872.117 |
| 17 | 140.814 | 4.97975 | 92.3759 | 817.420 |
| 18 | 142.297 | 4.96476 | 92.0510 | 768.855 |
| 19 | 143.763 | 4.95407 | 91.8065 | 725.701 |
| 20 | 145.219 | 4.94823 | 91.6526 | 687.129 |
| 21 | 146.643 | 4.94385 | 91.5268 | 652.535 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 108 Reynolds Number = 43313.6 Max. SPL = .0 DB
$T_o$ = 110.83 F $T_s$ = 211.53 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 112.830 | 8.85300 | 165.843 | 77478.4 |
| 2 | 114.068 | 6.30134 | 117.910 | 17774.1 |
| 3 | 116.352 | 5.90777 | 110.410 | 9232.27 |
| 4 | 118.468 | 5.53255 | 103.240 | 5613.77 |
| 5 | 120.500 | 5.49966 | 102.479 | 4030.18 |
| 6 | 122.415 | 5.29756 | 98.5791 | 3145.98 |
| 7 | 124.275 | 5.23075 | 97.2092 | 2577.64 |
| 8 | 126.096 | 5.18421 | 96.2224 | 2179.16 |
| 9 | 127.869 | 5.28829 | 98.0331 | 1890.52 |
| 10 | 129.617 | 5.10452 | 94.5121 | 1666.70 |
| 11 | 131.288 | 5.14947 | 95.2329 | 1476.35 |
| 12 | 132.930 | 5.12937 | 94.7538 | 1347.31 |
| 13 | 134.543 | 5.12068 | 94.4882 | 1229.26 |
| 14 | 136.117 | 5.10003 | 94.0050 | 1129.82 |
| 15 | 137.660 | 5.09860 | 93.8789 | 1045.09 |
| 16 | 139.170 | 5.13752 | 94.4975 | 972.326 |
| 17 | 140.665 | 5.17396 | 95.0708 | 909.063 |
| 18 | 142.137 | 5.17307 | 94.9590 | 853.199 |
| 19 | 143.583 | 5.18993 | 95.1743 | 803.578 |
| 20 | 145.014 | 5.24875 | 96.1596 | 759.357 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 112.830 | 8.85331 | 165.848 | 26012.8 |
| 2 | 114.068 | 7.57861 | 141.908 | 13652.0 |
| 3 | 116.352 | 6.73760 | 126.060 | 7025.93 |
| 4 | 118.468 | 6.32100 | 118.179 | 4708.42 |
| 5 | 120.500 | 6.10495 | 114.059 | 3550.41 |
| 6 | 122.415 | 5.93464 | 110.804 | 2848.90 |
| 7 | 124.275 | 5.80886 | 108.385 | 2376.42 |
| 8 | 126.096 | 5.71177 | 106.506 | 2035.59 |
| 9 | 127.869 | 5.65282 | 105.343 | 1785.56 |
| 10 | 129.617 | 5.58469 | 104.010 | 1582.72 |
| 11 | 131.288 | 5.53628 | 103.049 | 1425.74 |
| 12 | 132.930 | 5.49455 | 102.214 | 1296.18 |
| 13 | 134.543 | 5.45889 | 101.495 | 1187.60 |
| 14 | 136.117 | 5.42706 | 100.848 | 1095.73 |
| 15 | 137.660 | 5.39965 | 100.285 | 1016.93 |
| 16 | 139.170 | 5.37867 | 99.8443 | 949.182 |
| 17 | 140.665 | 5.36249 | 99.4928 | 889.651 |
| 18 | 142.137 | 5.34803 | 99.1744 | 836.798 |
| 19 | 143.583 | 5.33610 | 98.9041 | 789.836 |
| 20 | 145.014 | 5.32856 | 98.7159 | 747.861 |
| 21 | 147.021 | 5.43671 | 100.650 | 710.067 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 101 Reynolds Number = 46501.9 Max. SPL = .0 DB
$T_o$ = 110.53 F $T_s$ = 211.58 F Frequency = .0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 112.486 | 9.24226 | 173.167 | 83188.8 |
| 2 | 113.710 | 6.66279 | 124.699 | 19084.5 |
| 3 | 115.959 | 6.21853 | 116.244 | 9913.08 |
| 4 | 118.042 | 5.81733 | 108.581 | 6027.89 |
| 5 | 120.026 | 5.73304 | 106.857 | 4327.60 |
| 6 | 121.919 | 5.58865 | 104.026 | 3378.23 |
| 7 | 123.751 | 5.49474 | 102.147 | 2767.97 |
| 8 | 125.549 | 5.45931 | 101.362 | 2340.11 |
| 9 | 127.284 | 5.51459 | 102.264 | 2030.19 |
| 10 | 128.987 | 5.29557 | 98.0869 | 1789.88 |
| 11 | 130.644 | 5.43716 | 100.593 | 1585.49 |
| 12 | 132.258 | 5.36338 | 99.1177 | 1446.94 |
| 13 | 133.853 | 5.38283 | 99.3682 | 1320.18 |
| 14 | 135.402 | 5.33468 | 98.3737 | 1213.40 |
| 15 | 136.914 | 5.31120 | 97.8388 | 1122.43 |
| 16 | 138.407 | 5.39204 | 99.2264 | 1044.29 |
| 17 | 139.882 | 5.41968 | 99.6347 | 976.359 |
| 18 | 141.326 | 5.38218 | 98.8474 | 916.373 |
| 19 | 142.756 | 5.43664 | 99.7505 | 863.090 |
| 20 | 144.167 | 5.48429 | 100.528 | 815.605 |
| 21 | 145.560 | 5.54590 | 101.561 | 773.092 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 112.486 | 9.24256 | 173.173 | 27930.0 |
| 2 | 113.710 | 7.95108 | 148.911 | 14658.3 |
| 3 | 115.959 | 7.07852 | 132.466 | 7543.89 |
| 4 | 118.042 | 6.64242 | 124.215 | 5055.59 |
| 5 | 120.026 | 6.40414 | 119.677 | 3812.26 |
| 6 | 121.919 | 6.23161 | 116.376 | 3059.03 |
| 7 | 123.751 | 6.09990 | 113.844 | 2551.73 |
| 8 | 125.549 | 6.00012 | 111.912 | 2185.77 |
| 9 | 127.284 | 5.93327 | 110.599 | 1917.32 |
| 10 | 128.987 | 5.85480 | 109.072 | 1699.53 |
| 11 | 130.644 | 5.80782 | 108.135 | 1530.98 |
| 12 | 132.258 | 5.76248 | 107.231 | 1391.86 |
| 13 | 133.853 | 5.72610 | 106.495 | 1275.28 |
| 14 | 135.402 | 5.69157 | 105.796 | 1176.64 |
| 15 | 136.914 | 5.66029 | 105.160 | 1092.03 |
| 16 | 138.407 | 5.63872 | 104.706 | 1019.28 |
| 17 | 139.882 | 5.62149 | 104.333 | 955.363 |
| 18 | 141.326 | 5.60395 | 103.956 | 898.614 |
| 19 | 142.756 | 5.59137 | 103.671 | 848.187 |
| 20 | 144.167 | 5.58265 | 103.460 | 803.117 |
| 21 | 145.560 | 5.57782 | 103.321 | 762.691 |

## SUMMARY TABULATION OF LOCAL DATA

Run Number = 110 Reynolds Number = 9735.5 Max. SPL = 154.0 DB
$T_O$ = 110.97 F $T_s$ = 211.93 F Frequency = 227.0 CPS

| Chamber | T | h | Nu | Gz |
|---|---|---|---|---|
| 1 | 114.501 | 3.52915 | 66.0513 | 17385.1 |
| 2 | 116.407 | 2.21528 | 41.3829 | 3985.25 |
| 3 | 119.695 | 1.95952 | 36.5394 | 2068.83 |
| 4 | 122.490 | 1.70064 | 31.6453 | 1257.24 |
| 5 | 125.016 | 1.60319 | 29.7774 | 902.232 |
| 6 | 127.708 | 1.75994 | 32.6303 | 703.983 |
| 7 | 130.486 | 1.86792 | 34.5674 | 576.473 |
| 8 | 133.107 | 1.80595 | 33.3589 | 487.074 |
| 9 | 135.454 | 1.71142 | 31.5593 | 422.364 |
| 10 | 137.689 | 1.61040 | 29.6500 | 372.228 |
| 11 | 139914 | 1.70742 | 31.3888 | 329.603 |
| 12 | 142.258 | 1.84380 | 33.8435 | 300.670 |
| 13 | 144.580 | 1.87929 | 34.4406 | 274.197 |
| 14 | 146.609 | 1.69568 | 31.0301 | 251.920 |
| 15 | 148.568 | 1.68647 | 30.8201 | 232.963 |
| 16 | 150.544 | 1.76960 | 32.2965 | 216.679 |
| 17 | 152.570 | 1.86879 | 34.0611 | 202.516 |
| 18 | 154.629 | 1.95334 | 35.5533 | 190.002 |
| 19 | 156.593 | 1.92983 | 35.0783 | 178.887 |
| 20 | 158.472 | 1.91276 | 34.7234 | 168.991 |

## SUMMARY TABULATION OF AVERAGE DATA

| | | | | |
|---|---|---|---|---|
| 1 | 114.501 | 3.52952 | 66.0582 | 5836.93 |
| 2 | 116.407 | 2.88334 | 53.9287 | 3062.63 |
| 3 | 119.695 | 2.42520 | 45.3080 | 1575.61 |
| 4 | 122.490 | 2.18052 | 40.6974 | 1055.65 |
| 5 | 125.016 | 2.03462 | 37.9412 | 795.884 |
| 6 | 127.708 | 1.97679 | 36.8284 | 638.458 |
| 7 | 130.486 | 1.95545 | 36.3959 | 532.404 |
| 8 | 133.107 | 1.93137 | 35.9152 | 455.921 |
| 9 | 135.454 | 1.90216 | 35.3435 | 399.843 |
| 10 | 137.689 | 1.86724 | 34.6680 | 354.362 |
| 11 | 139.914 | 1.84967 | 34.3154 | 319.156 |
| 12 | 142.258 | 1.84736 | 34.2451 | 290.083 |
| 13 | 144.580 | 1.84836 | 34.2364 | 265.720 |
| 14 | 146.609 | 1.83522 | 33.9693 | 245.128 |
| 15 | 148.568 | 1.82330 | 33.7260 | 227.467 |
| 16 | 150.544 | 1.81849 | 33.6145 | 212.279 |
| 17 | 152.570 | 1.82039 | 33.6262 | 198.930 |
| 18 | 154.629 | 1.82699 | 33.7243 | 187.075 |
| 19 | 156.593 | 1.83155 | 33.7860 | 176.545 |
| 20 | 158.472 | 1.83477 | 33.8236 | 167.138 |

# APPENDIX B

## CHAMBER DIMENSIONS

| Chamber Number | $x_i$ (ft.) | $A_{xi}$ (ft.$^2$) | $A_i$ (ft.$^2$) |
|---|---|---|---|
| 1 | 0.267 | 0.279 | 0.279 |
| 2 | 0.527 | 0.247 | 0.532 |
| 3 | 1.023 | 0.495 | 1.033 |
| 4 | 1.526 | 0.502 | 1.540 |
| 5 | 2.022 | 0.496 | 2.041 |
| 6 | 2.518 | 0.496 | 2.542 |
| 7 | 3.017 | 0.498 | 3.046 |
| 8 | 3.520 | 0.502 | 3.554 |
| 9 | 4.010 | 0.490 | 4.049 |
| 10 | 4.521 | 0.511 | 4.565 |
| 11 | 5.016 | 0.494 | 5.065 |
| 12 | 5.514 | 0.498 | 5.568 |
| 13 | 6.015 | 0.500 | 6.974 |
| 14 | 6.516 | 0.500 | 6.579 |
| 15 | 7.017 | 0.501 | 7.085 |
| 16 | 7.514 | 0.496 | 7.587 |
| 17 | 8.013 | 0.498 | 8.091 |
| 18 | 8.515 | 0.501 | 8.597 |
| 19 | 9.017 | 0.501 | 9.104 |
| 20 | 9.518 | 0.501 | 6.610 |
| 21 | 10.018 | 0.499 | 10.115 |

# LITERATURE CITED

1. McAdams, W. H., Heat Transmission, McGraw-Hill Book Company, Inc., New York, Pages 219 and 225, 3rd Edition, 1954.

2. Eastwood, Ian, Threshold Values for Resonant Acoustic Vibrations in a Horizontal, Isothermal Tube, Unpublished MS Thesis, Georgia Institute of Technology, 1961.

3. Jackson, T. W., Spurlock, J. M., Purdy, K. R., Oliver, C. C. and Johnson, H. L., The Effects of Resonant Acoustic Vibrations on Heat Transfer to Air in Horizontal Tubes, WADC Technical Note 59-330, Wright Air Development Center, 1959.

4. Purdy, K. R., Viscous Fluid Flow Under the Influence of a Resonant Acoustic Field, Unpublished PhD Thesis, Georgia Institute of Technology, 1963.

5. Schlichting, Hermann, Boundary Layer Theory, McGraw-Hill Book Company, Inc., New York, Page 502, 4th Edition, 1960.

6. Keenan, J. H., and Keyes, F. G., Thermodynamic Properties of Steam, First Edition, J. Wiley and Sons, New York, 1936.

7. Hilsenrath, J., Tables of Thermal Properties of Gases, National Bureau of Standards Circular 564, Washington, D. C., 1955.

8. Boelter, L. M. K., Young, G., Iverson, H. W., Distribution of Heat Transfer Rate in the Entrance Section of a Circular Tube, NACA TN 1451, Washington, 1948.